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DEVELOPMENT OF MEASUREMENT TECHNIQUES FOR THE ANALYSIS OF TRACKED VEHICLE VIBRATION AND NOISE

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ABSTRACT

A comprehensive study has been made of the noise and vibration characteristics of the M-113 Armored Personnel Carrier. As the initial and primary phase of this program, measurement techniques and instrumentation systems were evolved to define vibratory energy sources, transmission paths, and noise radiating sources of the vehicle. The measurement techniques have been evolved for use by design personnel concerned with virtually any type of tracked vehicle. While instrumentation systems are simple and straightforward, when used with the systematic test sequence evolved, they serve to effectively evaluate noise offenders both within and outside the vehicle.

Results of the tests show that noise conditions within the M-113 vehicle are above the limits prescribed by most hearing damage criteria, at virtually all normal operating conditions. The track-sprocket wheel reaction and the track-idler wheel reaction have been shown to be the chief sources of vibration, while predominant interior radiators vary with operating conditions. Important radiators include floor panels, engine access panels, and the variety of other thin metallic panels or components within the vehicle.

In addition to the primary measurement and analysis work, some consideration has been given to treatment and redesign techniques which might be used to ameliorate existing noise conditions in the M-113. In conclusion, recommendations are given for further activity in the area of tracked vehicle noise control.

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I. INTRODUCTION

The objective of the program discussed in this report was to develop noise and vibration measurement techniques suitable for the identification and evaluation of the sources of vibration existing in tracked vehicles, the sources of radiated noise, and the transmission paths between the vibration sources and noise radiators. Secondary objectives have been to study the nature and severity of the generated vibrations and radiated noise created by tracked vehicles, and to recommend general approaches for subduing these problems.

During the course of this program a detailed study was made of track and suspension system noise and vibration on an M-113 Armored Personnel Carrier. Although the measurement techniques developed were specifically used on this vehicle, they are readily adaptable to tracked military vehicles in general.

From available literature on noise and vibration in tracked vehicles, it was apparent that several earlier programs have been conducted on reducing the noise levels in tracked vehicles, and specifically on the M-113. A survey of available reports[1] shows that extensive noise surveys have been conducted and some work has been concerned with treatment of the vehicle. This treatment has involved both the modification of radiating sources and interior absorption treatments; however, the results achieved have been marginal and largely unpredictable. Their results show that some attempts have been successful in reducing noise levels while others produced relatively insignificant effects.

Thus, it became apparent that in order to make any significant advances in the reduction of noise and vibration conditions in tracked vehicles, an efficiently developed systematic approach was needed. The logical first step in this approach has been the development of predictable measurement and analysis techniques. Before noise conditions could be effectively treated it was necessary to define the vibration and sound sources and evaluate the relative importance of each. For example, even though the treatment of one radiating source is successful in reducing the contribution from this particular source, the effect on the over-all conditions may be negligible if more important sources exist. The efforts of this program have been to provide that initial step suitable for defining these sources of vibration and sound so that an effective treatment procedure can be followed.

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1 Listed in Bibliography

II. INSTRUMENTATION

In order to evolve a measurement system and technique best suited to the definition of tracked vehicle noise and vibration, a variety of instrumentation systems were evaluated. It was recognized that at least three basic measuring techniques would have to be developed to totally define the system (1) for defining and evaluating the sources of vibratory energy in the track and suspension system, (2) for defining the paths by which solid-borne noise is conducted into and throughout the vehicle, and (3) for defining the radiators of acoustic noise within the vehicle. Also, there are certain prerequisites of any type of measuring system that can be developed. Some of these are (1) the capability of continuously recording any or all the pertinent variables, (2) an accuracy within about one decibel, (3) a flat frequency response from essentially dc throughout the audio range in order to accurately define both low frequency vibrations and acoustic noise inherent in the operation of tracked vehicles, (4) portability and (5) ruggedness to withstand the noise, shock, and vibration environment of the vehicle. Thus the final goal of the program was to evolve techniques which would be sufficiently detailed to cover the areas of interest, but yet be simple and rugged enough for effective use by design and test groups.

In order to impart better understanding of the data discussed in this report, all the basic instrumentation systems used in the program are discussed in some detail in the text of the report. Final recommendations on instrumentation systems and measuring techniques are given in Section IV. The bulk of vibration and sound data were recorded with systems involving either a light beam oscillograph or a magnetic tape recorder. Using these two recording instruments with appropriate instrumentation systems, all the required aspects were satisfactorily met insofar as measurement and recording were concerned. For example, the oscillograph recording system was adaptable to low frequency and was capable of furnishing a continuous record of sound level and vibration waveform as a function of operating conditions, thus making it possible to correlate resonances with operating conditions. The tape recording system was invaluable for providing data since complete spectral analysis, octave band analysis, 1/3 octave band analysis, and narrow band analysis could all be obtained from the same data loop. The details of these two systems and the other instrumentation used are given in the following paragraphs.

A. CEC Oscillograph Recordings

Numerous vibration and sound surveys were conducted with the basic instrumentation system shown in Figure 1. To measure vibrations, five light weight Endevco Model 2215C wide band piezo-electric accelerometers were used. These were provided with matched miniature cables. The electrical signal from these transducers was received by five special transistorized amplifiers of 22 megohms input impedance, with pass band of 2 to 10,000 cps, gains of 10 to 1000 in 5 steps, and outputs of up to 2 volts across 82 ohms. These were specially designed and built at SwRI for this program. The output of the amplifiers was used to drive Type 7-362 (2500 cps upper cut-off) or Type 7-361 (5000 cps upper cut-off) galvanometers in a Consolidated Electrodynamics Corporation Type 5-116, 14 channel, light beam oscillograph.

The accelerometers were rigidly attached to the surfaces of the vehicle by screwing their studs into small aluminum blocks bonded to the surfaces under investigation. The transducer and block together weigh only 75 grams, and do not appreciably alter the vibration of the large metal panels to which they are applied.

The intensity of sound radiation from a solid surface is proportional to the surface velocity, rather than its acceleration[2], therefore, the accelerometer signals were electronically integrated in order that the velocity could be read directly. The outputs of the amplifiers are proportional to the vibrational velocities of the pickups for frequencies above about 10 cps.

Sound was recorded by this system employing the light beam oscillograph by using a Shure Model 98B99 ceramic microphone, Electro Model A-15 dc amplifier, and a type 7-361 galvonometer in the CEC recorder.

This method of vibration and sound measurement was extremely valuable for analyzing the vibrational character of the vehicle as a function of operating conditions, since a continuous record was obtained as the conditions were varied. In order to more directly correlate the measured vibrations to operating conditions, the engine rpm and vehicle speed were continuously recorded on the CEC chart paper, along with the vibration and sound records. These records were produced by two tachometer generators connected to the rpm indicator shaft and speedometer shaft by means of duplexer fittings and flexible cables. The signal from the generators yielded sine-waves on the CEC records whose frequencies were proportional to rpm and track speed, respectively. Thus, continuous records of these two important vibration-exciting frequencies were afforded, regardless of variations in the vehicle's automatic transmission ratio.

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2 See Section III-C

B. Magnetic Tape Recordings

1. Data Recording Instrumentation

A second method of data taking which proved invaluable was that using a portable magnetic tape recorder. By making tape recordings of the vibration and sound data, it was possible to conduct frequency analyses to the extent desired. Sound data was taken using the Shure Model 98B99 ceramic microphone with an Ampex Model 1260 tape recorder. Vibration measurements were recorded using the Endevco Model 2215C accelerometers and the transistorized amplifiers described in the previous section with the recorder.

2. Analyzing Instrumentation

The complete measuring and analyzing systems used for the taped data are illustrated in Figure 2. Once the data were recorded, selected portions of the recordings were made into continuous loops for detailed analyses. Several types of analyzers were used, depending upon the objectives of the particular analysis. Because of the wide-band nature of the noise levels present, a General Radio Type 1550-A Octave Band Analyzer was used for most of the analyses.

For more detailed analyses of the spectra, a Panoramic Radio Products Company Type LF-2 subsonic analyzer was used. The frequency range of the analyzer, from 1/2 cps to 2000 cps, was sufficient for the typical vibration and noise spectra observed. The spectra were analyzed in intervals of 500 cps (i.e., 1/2 cps to 500 cps, 500 cps to 1000 cps, etc.). Due to the random nature of the spectra, it was necessary to use a slow scan time (16 min.) to obtain repeatable results. A logarithmic amplitude scale was used so that a greater range in levels could be observed.

For high frequency analyses, the tape loops were replayed into a Panoramic Radio Products Company Type LP-1a sound analyzer. The spectra were observed on the display screen of the analyzer, and photographs were made of noticeable high frequency components.

The frequency response of the tape recording system was essentially flat throughout the range of interest except at frequencies below 50 cps. Down to 50 cps the response was found to be within 1 db; however, the response was down 5 db at 40 cps. Thus, at low frequency it was necessary to refer to CEC oscillograph recordings to supplement the taped data.

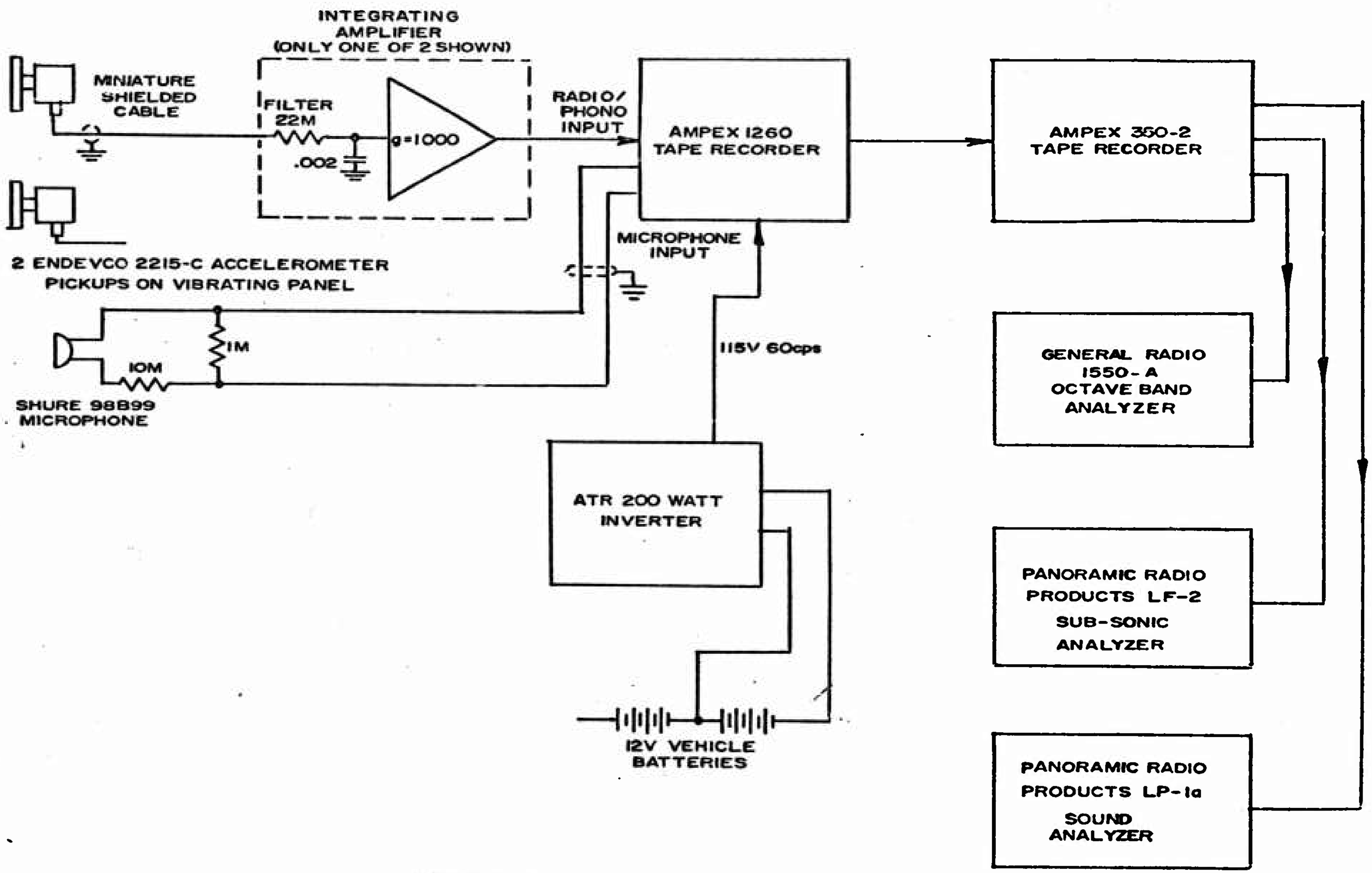

FIGURE 2
MAGNETIC TAPE RECORDER INSTRUMENTATION SYSTEM INCLUDING ANALYZERS

C. Other Instrumentation

Another instrument used extensively was the General Radio Type 1551-B Sound Level Meter. This instrument was used throughout the testing period for making quick noise level surveys and for providing data for checking the reliability of our other noise measuring systems.

A General Radio Type 1556-B Impact Noise Analyzer was used in conjunction with the sound level meter for measuring impact noise levels. Impacts were delivered to various vibration sources and measurement of the noise produced gave some insight into the relative transmission between the vibration sources and the noise emitters.

D. Power Supply and Calibrations

Power for the instruments requiring a 110V, 60 cps source was furnished by a ATR 200-watt inverter driven by one of the vehicle batteries. For the driveby tests a 12V storage battery was used.

To calibrate the various instruments, a General Radio Type 1552-B Sound Level Calibrator driven by a General Radio Type 1307-A Transistor Oscillator, and a General Radio Type 1557-A Vibration Calibrator were used.

To insure against any undesired effects that might occur due to electronic drift, changing environmental conditions, etc., calibration signals were recorded each day that either the light beam oscillograph or the magnetic tape recorder were used for testing.

III. PROCEDURES, DATA, AND RESULTS

A. Summary of Testing Approach

As the first step in the experimental program, extensive noise and vibration surveys were made of the M-113 Armored Personnel Carrier. The objective of these initial tests was to gain a fundamental understanding of the existing vibration and noise environment under all operating conditions in order to fully define problem scope in terms of severity, spectrum, importance of operating conditions, etc. After the characteristics of the generated vibrations and radiated noise were defined, efforts thereafter were concentrated toward identifying vibration sources and tracing the transmission paths to the noise radiators. In addition, several general treatment techniques were evaluated during the course of the program both to define the effectiveness of these treatment approaches and to assist in source identification.

The following is a brief outline of the testing procedures as discussed in this report.

1. Noise Surveys

a. Preliminary

These were made to define in quantitative terms the severity of the noise conditions both inside and outside the vehicle. Primary emphasis was placed on interior conditions relative to hearing damage risk criteria for personnel.

b. Dependence Upon Operating Conditions

Internal and exterior noise conditions were measured and analyzed as a function of operating conditions. These served to define the important operating conditions for later detailed testing, and the sensitivity of noise conditions to changes in operating condition, test point location, and the like.

c. Source Identification

Acoustical tests inside and outside were also used to evaluate the importance of individual contributing sources (usually conducted in conjunction with vibration surveys).

By sequentially treating or removing radiating sources in the vehicle, it was possible to infer their contribution to the overall noise. Similarly, noise tests were made as modifications were made of the track and suspension system to evaluate the importance of various generation mechanisms and transmission paths into the vehicle.

2. Interior Vibration Surveys

Preliminary vibration surveys were conducted first to attest the adequacy of the instrumentation systems utilized, to define those operating conditions (speed, etc.) best suited to further study in the test program, and to study the correlation between vibration conditions and radiated noise within the vehicle.

Interior vibration testing during the major portion of the test program was then conducted to identify and evaluate the principle noise radiating surfaces within the vehicle. Also, interior tests made during selective modification of the track and suspension system helped define generation and transmission paths of the solid-borne noise.

Impact testing was also used to define important characteristics (resonant frequency, damping, etc.) of the interior noise radiators.

3. Vibration Source and Transmission Path Studies

This testing involved sound and vibration measurement both inside and outside the vehicles. As noted above, interior noise and vibration conditions helped define the importance of generation mechanisms and transmission paths, as the track and suspension system was systematically altered. In addition, vibration measurements were made on the suspension system itself, and modification of this system was then used to sequentially remove important sources and paths in order that resulting effects could be defined. Such testing was made under normal operating conditions and with the vehicle blocked up. Impact test techniques were also used to compare the importance of individual transmission paths.

4. Treatment Techniques

During the course of the testing program, acoustical treatments were made of individual noise radiators and internal environmental acoustical conditions in order to selectively remove individual noise sources and to reduce background levels for more effective testing of individual sources. In addition, several basic treatment techniques were evaluated relative to their potential and practicality in reducing existing noise conditions within the vehicle.

More detailed discussions of these test techniques, together with analyzing techniques used and results obtained are given in the remainder of this section.

B. Evaluation of Noise Characteristics

1. Noise Criteria

The first step in evaluating any suspected noise problem is, of course, the definition of effective criteria which may be used as a basis for quantitative comparison. Considerable data has been published in the literature concerning the effects of noise on personnel, and while it is not the purpose of this report to analyze and comment extensively on these data, it is of interest to present some of the published noise exposure criteria to serve as a basis for quantitatively evaluating noise conditions in the M-113 APC.

The most important effect of noise exposure on personnel is the threshold shift. The threshold shift can be temporary or permanent depending upon exposure time and noise levels. A temporary hearing loss or threshold shift is one that disappears after a period of time when the subject is removed from the sound field, and is due to the natural tendency of sensory organisms to protect themselves from overstimulation. When stimulation is sufficiently prolonged and/or intense that the ear does not completely recover then the ear is said to have sustained a permanent threshold shift or permanent hearing damage. In discussing the important factors related to permanent hearing damage the Sub-Committee on Noise of the Commmittee on Conservation of Hearing[3] gave these factors as most pertinent: (1) Many noise exposures can produce a permanent hearing loss that may affect communication by speech; (2) noise induced hearing loss may be temporary, permanent, or a combination of temporary and permanent; (3) permanent noise induced hearing loss is due to destruction of certain inner ear structure which cannot be replaced or repaired: (4) the amount of hearing loss produced by a given noise exposure varies from person to person; (5) noise induced hearing loss first affects man's hearing of sounds higher in frequency than those essential for communication by speech (therefore, most early noise induced hearing losses pass unnoticed unless they

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3 "Guide for the Conservation of Hearing in Noise", Prepared by Sub-Committee on Noise of the Committee on Conservation of Hearing and Research Center Sub-Committee on Noise, 1964.

are detected by suitable hearing tests); and (6) the four major factors which characterize noise exposure are over-all noise level, composition of the noise, duration and distribution of the exposure during a typical work day, and the total time of exposure during a work life.

In defining an adequate criteria for the allowable noise levels as a function of frequency for a military vehicle such as the M-113 APC, all of the factors mentioned previously have to be considered. Also there are other factors which must be considered, such as allowable temporary threshold shift, and less tangible effects such as physiological and psychological effects (noise induced fatigue, morale degradation, limited communication, annoyance, etc.).

Numerous attempts have been made to establish a Damage Risk (DR) Criterion of exposure limits below which a large portion of the population would not sustain significant permanent hearing loss. Several of these Damage Risk Criteria evolved are shown in Figure 3. These criteria are based upon octave band pressure levels which have, in general, been evolved for particular exposure durations. For example, Curve A and Curve B (for pure tone and wide band noise respectively) are based upon an exposure duration of 40 hours per week for life. Curve C is from MIL S1-10, "General Specifications for Ships of the US Navy", and is designed for the normal exposure to which personnel are subjected while on watch in shipboard machinery spaces. Curve D is a standard from the Army Human Engineering Laboratories: HEL S-1-63, "Maximum Acceptable Noise Levels For Army Materiel Command Equipment". This is, of course, the standard to which noise conditions in the M-113 APC should be compared, and presumably is based upon the exposure duration to which personnel would normally be subjected.

In addition to these damage risk criteria, mention should be made of a more complex and perhaps more comprehensive approach to the problem of noise induced hearing loss made by Glorig et al[4]. The referenced paper provides a basis for evolving allowable exposure times for the 600-1200 cps band. The suggested criteria is that if no more than 12 db of temporary threshold shift is experienced at 2000 cps at the end of an exposure period and the TTS is allowed to recover, there will be no significant permanent threshold shift at the end of 10 years. Glorig's criteria for this

4 A. Glorig, W. D. Ward, J. Nixon, "Damage Risk Criteria and Noise Induced Hearing Loss", Archives of Otolaryngology, Vol 74, pp 413-423 (Oct 1961)

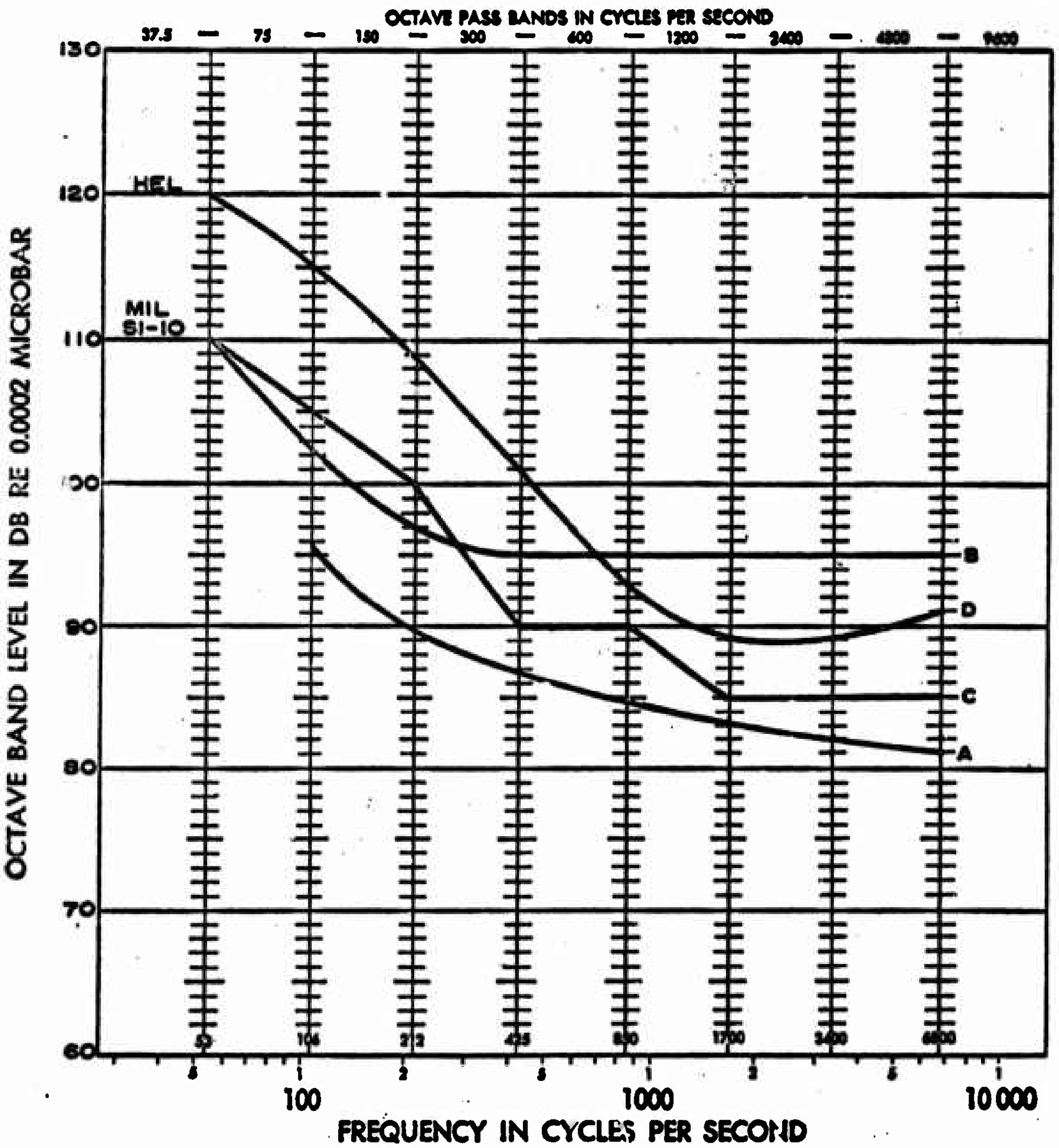


FIGURE 3
TYPICAL CRITERIA FOR PROTECTION OF PERSONNEL FROM NOISE INDUCED PERMANENT HEARING LOSS

was that if there is no threshold shift at 2000 cps at the end of 10 years then the only hearing loss would be that due to presbycusis.

2. Test Conditions

Most of the noise data for studying the characteristics of the M-113 tracked vehicle noise levels were taken with the sound level meter and the tape recorder system described in Section II-B. Overall levels were measured with the sound level meter, and spectral analyses were made from the tape data. A position at ear level near the center of the passenger compartment was selected and used as a standard for making inside noise level comparisons as operating conditions were varied throughout the testing period. This location was approximately 9" to the rear of the center post, and 10" below the ceiling. This central position was chosen because it was approximately equidistant from the various noise radiating panels, thus localized effects produced by these panels are minimized. For external noise level comparisons, a position about 16" above the ground and 50 feet from the vehicle was used in most instances.

Ambient noise levels ranged from about 50 db inside and 60 db outside the vehicle with all equipment off to nearly 80 db both inside and outside with the ATR inverter power supply running. However, these ambient noise levels did not seriously interfere with the sound levels created by the vehicle, and any effect could be conveniently compensated.

The test course used for most of the testing in this program was an old airport runway about 1.2 miles long. The data in this report was taken on this course unless otherwise noted. Initially all types of terrain were used to evaluate the effects of road surface on noise. Once the variation of noise with terrain was determined, the runway course was adopted for the tests studying other effects. This track was long, straight, level, and possessed a fairly consistent surface of gravel and crumbled pavement.

3. Survey of Internal Noise Conditions

a. Engine Idling Noise

Sound level studies inside the vehicle were first made to correlate the effects of operating conditions on resulting noise levels. The first tests were made to study the noise levels produced by the engine under idling conditions. The noise inside the vehicle with all hatches closed at engine idling speeds of 700 rpm and 2500 rpm are given in Figures 4 and 5.[5] It may

[5] These initial surveys were made with the periscope ports unsealed. The remainder of the results are with these ports closed.

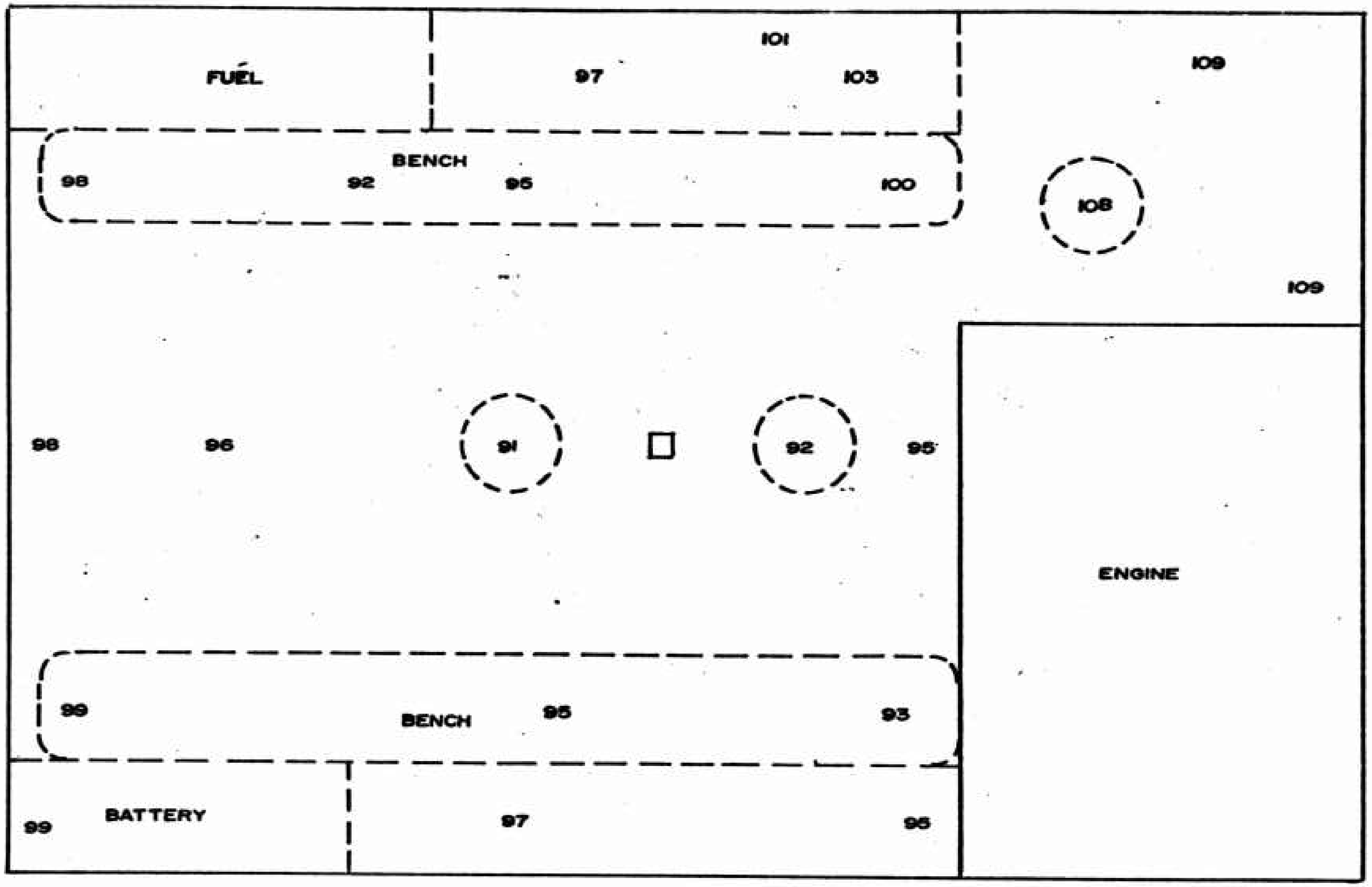


FIGURE 4

SOUND LEVEL SURVEY, DB, WITH ENGINE IDLING AT 700 RPM, ALL HATCHES CLOSED
(Measurements made with C weighting of sound level meter)

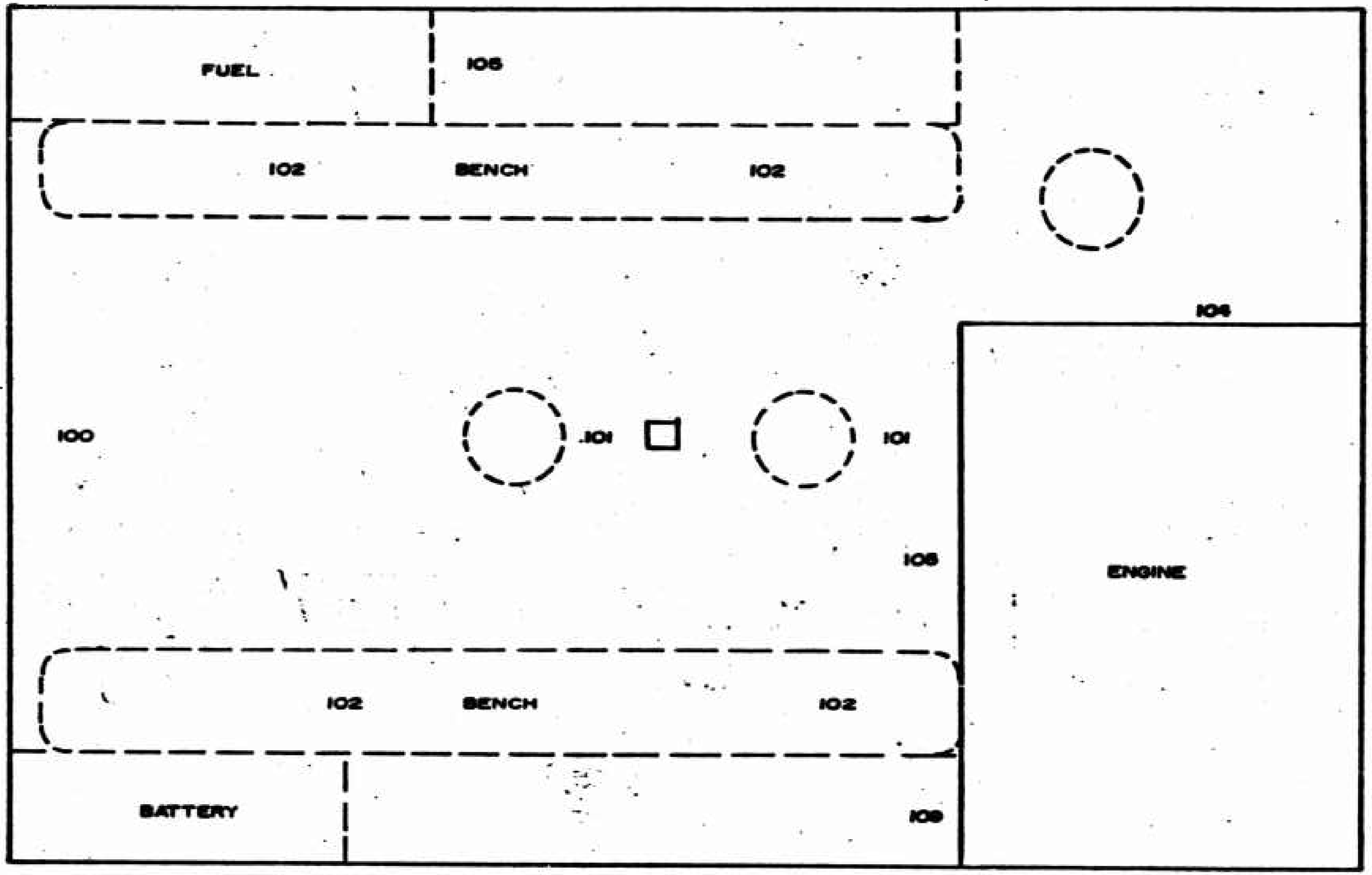


FIGURE 5. SOUND LEVEL SURVEY, DB, WITH ENGINE IDLING AT 2500 RPM, ALL HATCHES CLOSED

(Measurements made with C weighting of sound level meter)

be seen that sound levels increased from 2 to 10 db at various positions as the engine speed was increased from 700 to 2500 rpm. Since the noise levels near the walls are higher than at interior positions, it is indicated that these surfaces are either sound sources or antinodes in standing-wave sound-pressure patterns for various frequencies, or both, as one would expect. Also, it is noted that at low engine speeds, the noise levels were several decibels higher in the driver's compartment than elsewhere.

The overall noise level taken at the center of the passenger compartment as a function of engine speed and hatch position (open or closed) is given in the following table:

TABLE I. Variation of Internal Noise With Engine Idle Speed

Engine Speed, RPM	SPL, db All Hatches Open	SPL, db All Hatches Closed
750	89	96
1000	95	94
1500	94	95
2000	104	99
2500	99	97
3000	104	104

There was an increase of 15 db as the engine speed was increased with the hatches open and a 10 db increase with the hatches closed. There was a distinct resonance at 2000 rpm with the hatches both open and closed. Opening the hatches lowered the sound level at low speeds but tended to increase the levels at high speeds. It was evident during these tests that opening the hatches allows sound energy inside to escape at low speeds but lets energy in from the exhaust at higher speeds. Opening the rear ramp decreased the sound level by 4 to 10 db at all speeds.

Octave band analyses of the internal noise with the engine idling at 1500 rpm and 2500 rpm are given in Figure 6.

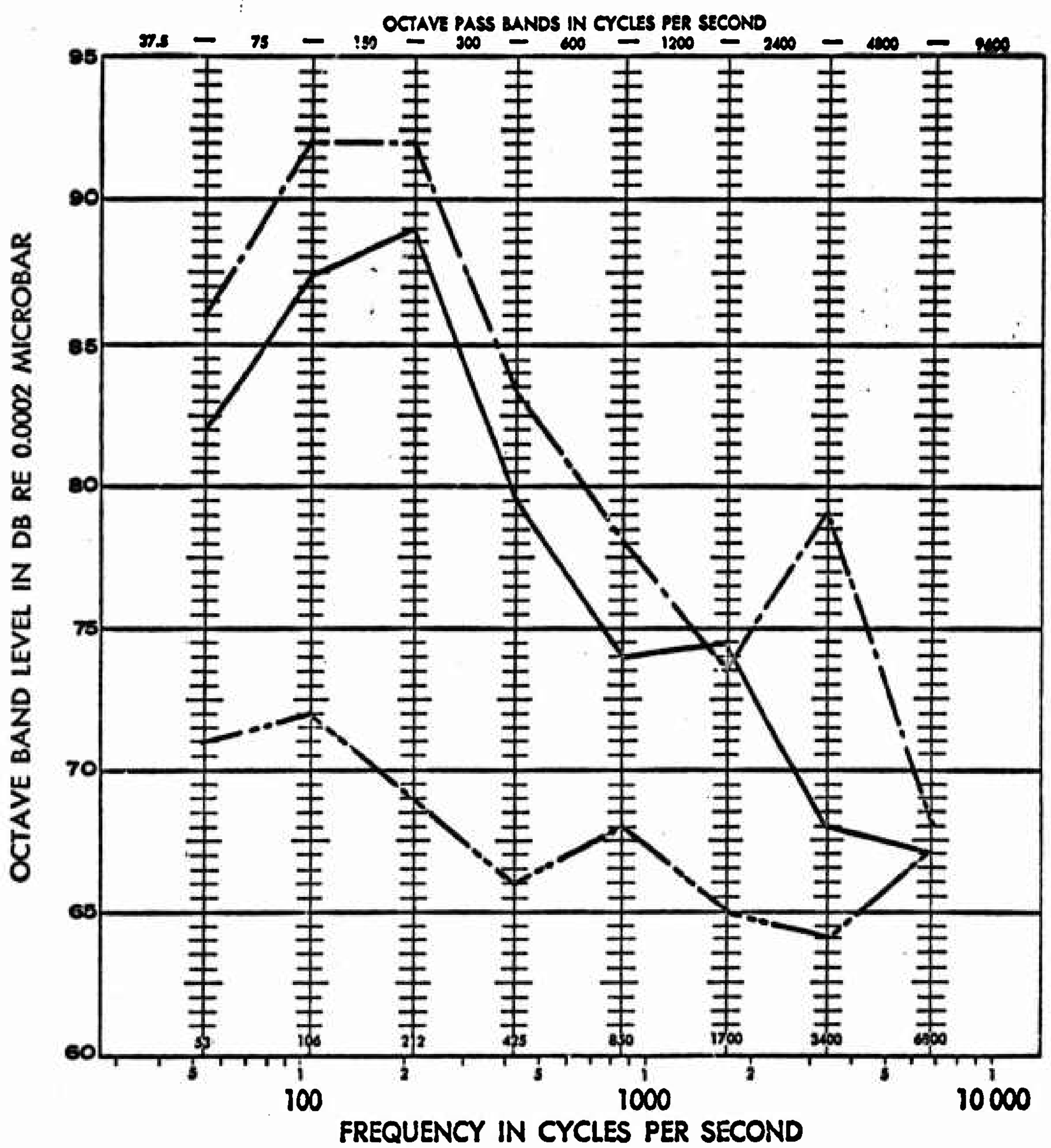
The engine noise was noted to vary considerably throughout the testing period, especially at low speeds. For example, an 8 db variation was observed in the noise level at 750 rpm from data taken on various days, in spite of regular re-calibration of equipment. This could be due to differences in the operating condition of the engine, or slight differences in the actual engine speed. It was noted that a difference of 100 rpm produced as much as 5 db difference in the noise level at various positions because of resonances and standing-wave patterns.

b. Noise Inside the Moving Vehicle

Next, tests were conducted to establish the nature and severity of the noise levels inside the moving vehicle and to study the effect of operating conditions on the observed levels. Sound measurements were made operating the vehicle at various speeds, over various terrains, up and down hill, and with the hatches both opened and closed.

Sound surveys at 10 mph and 20 mph on the standard test course illustrating the noise levels encountered by occupants are presented in Figures 7 and 8. The measurements were made at ear level at the indicated positions. The data shows that there is a variation of 5 to 6 db among the levels at the occupant positions at the same speed. The noisiest locations are that of the driver and the rear end of the right bench. The quietest position was in the center of the passenger compartment. Octave bands of the noise in these three positions are given in Figures 9 - 11. The severity of these noise levels are illustrated by comparison to the HEL standard.

Figures 7 - 11 also indicate the effect of the hatches being opened or closed on the sound level. At high speeds, opening the hatches reduces the noise levels by 1 to 2 db. At 10 mph, there are no consistent differences between the levels with the hatches open and closed.



1500 RPM
2500 RPM
AMBIENT NOISE

FIGURE 6. INTERNAL SOUND SPECTRA WITH ENGINE IDLING, ALL HATCHES CLOSED

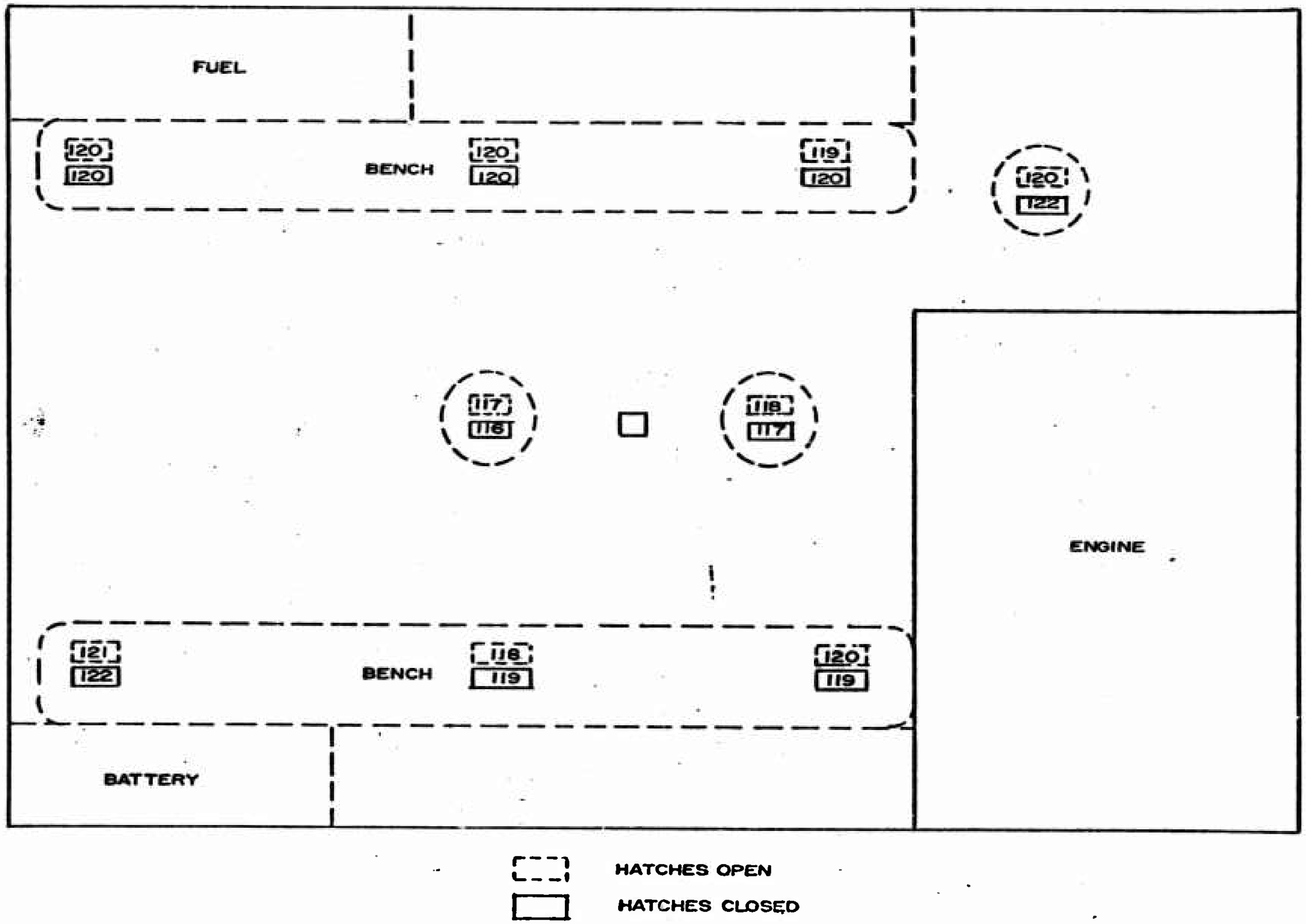


FIGURE 7. INTERNAL SOUND SURVEY, DB, AT 10 MPH
(Measurements made with C weighting of sound level meter)

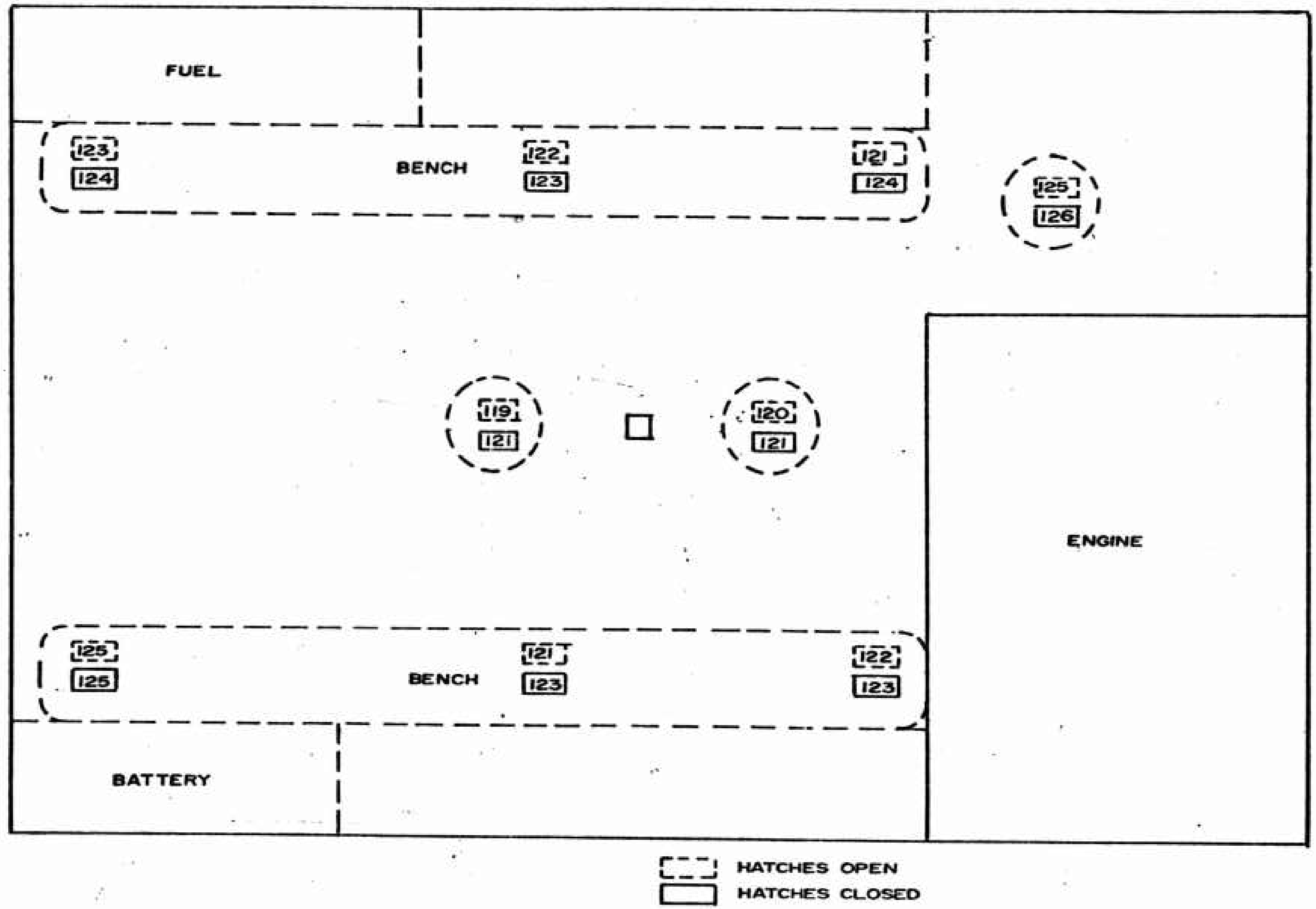
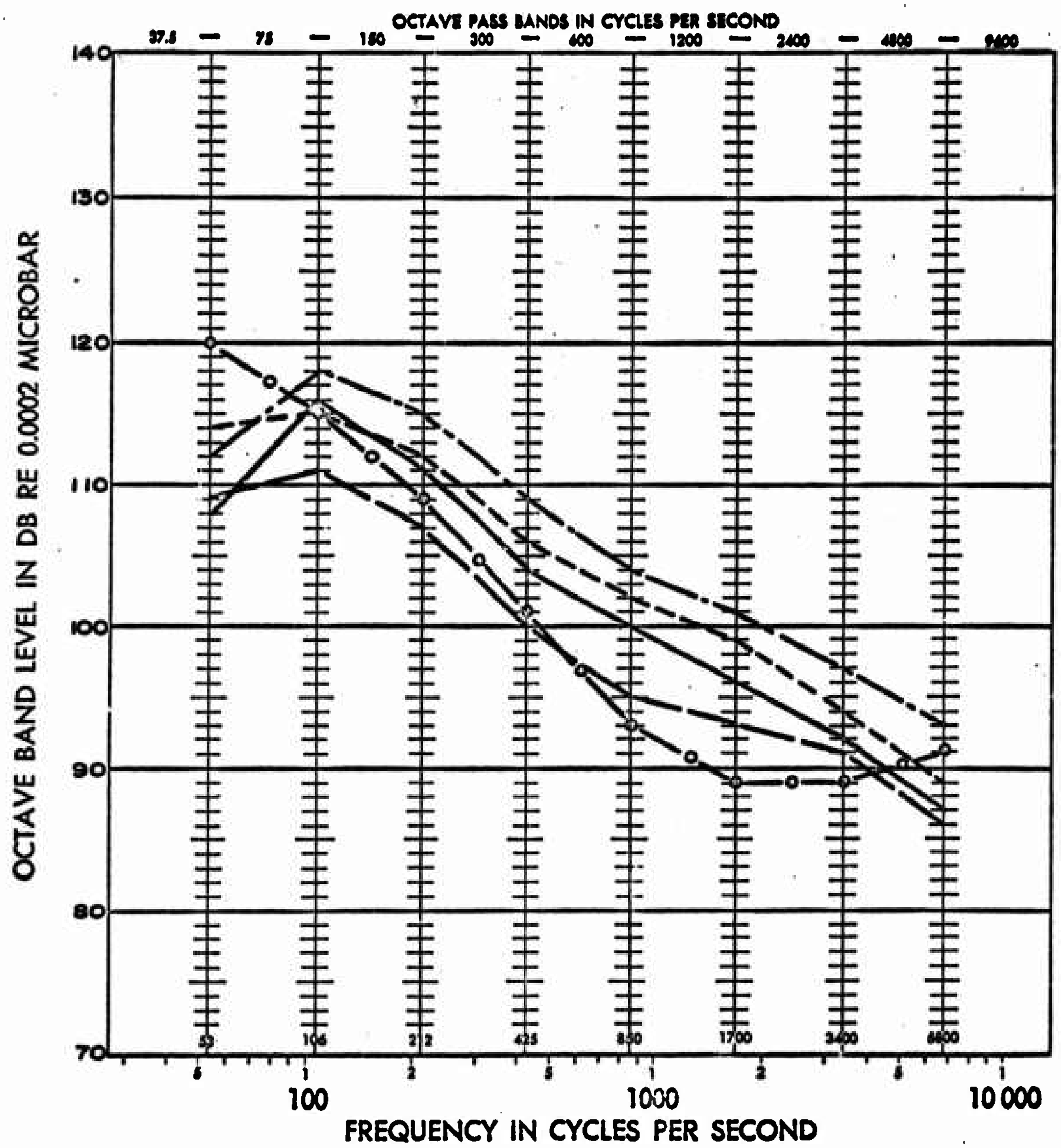


FIGURE 8. INTERNAL SOUND SURVEY, DB, AT 20 MPH.

(Measurements made with C weighting of sound level meter)



10 MPH, HATCHES OPEN
10 MPH, HATCHES CLOSED
20 MPH, HATCHES OPEN
20 MPH, HATCHES CLOSED
HEL STANDARD S-1-63

FIGURE 9. INTERNAL SOUND SPECTRA AT CENTER OF PASSENGER COMPARTMENT

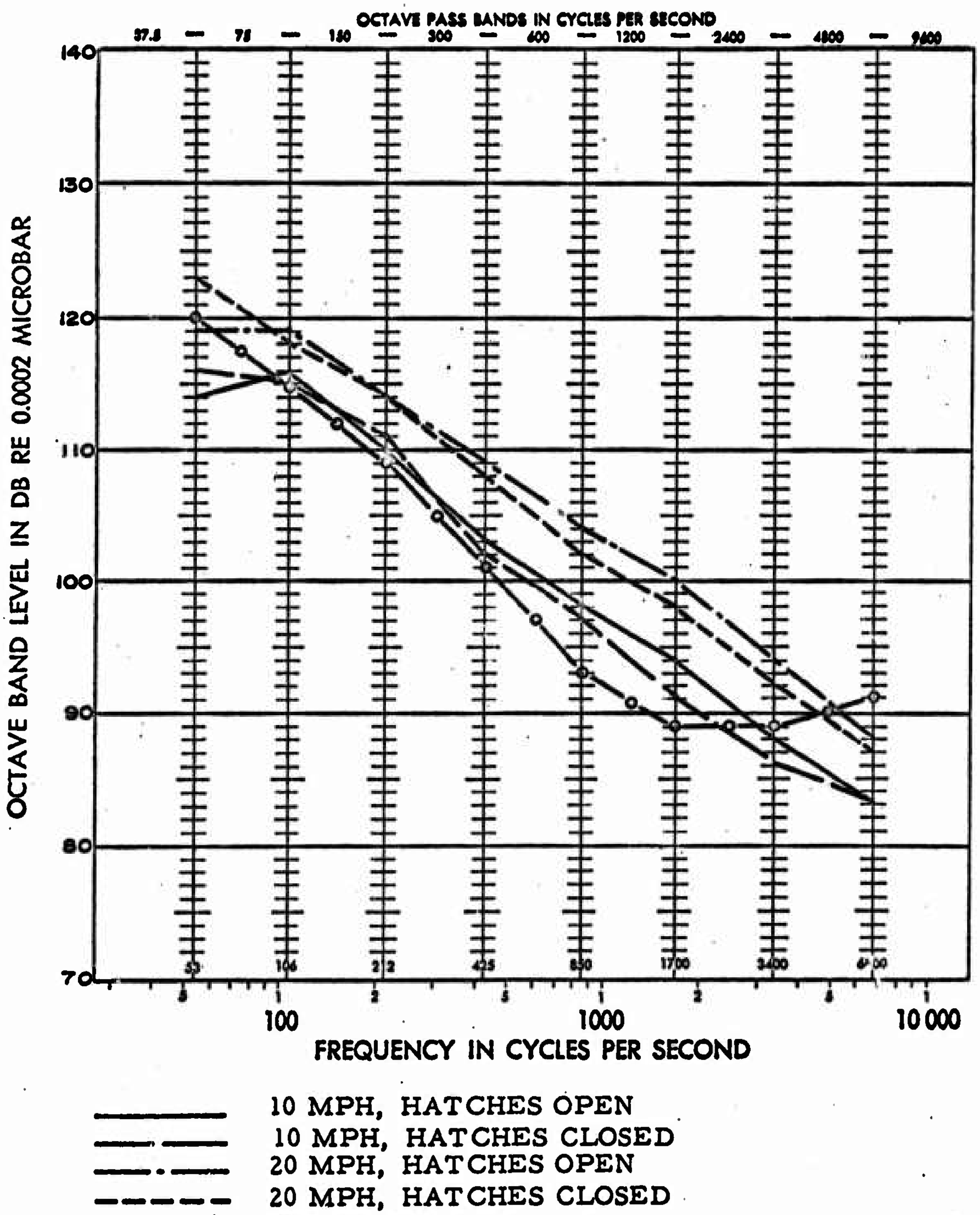
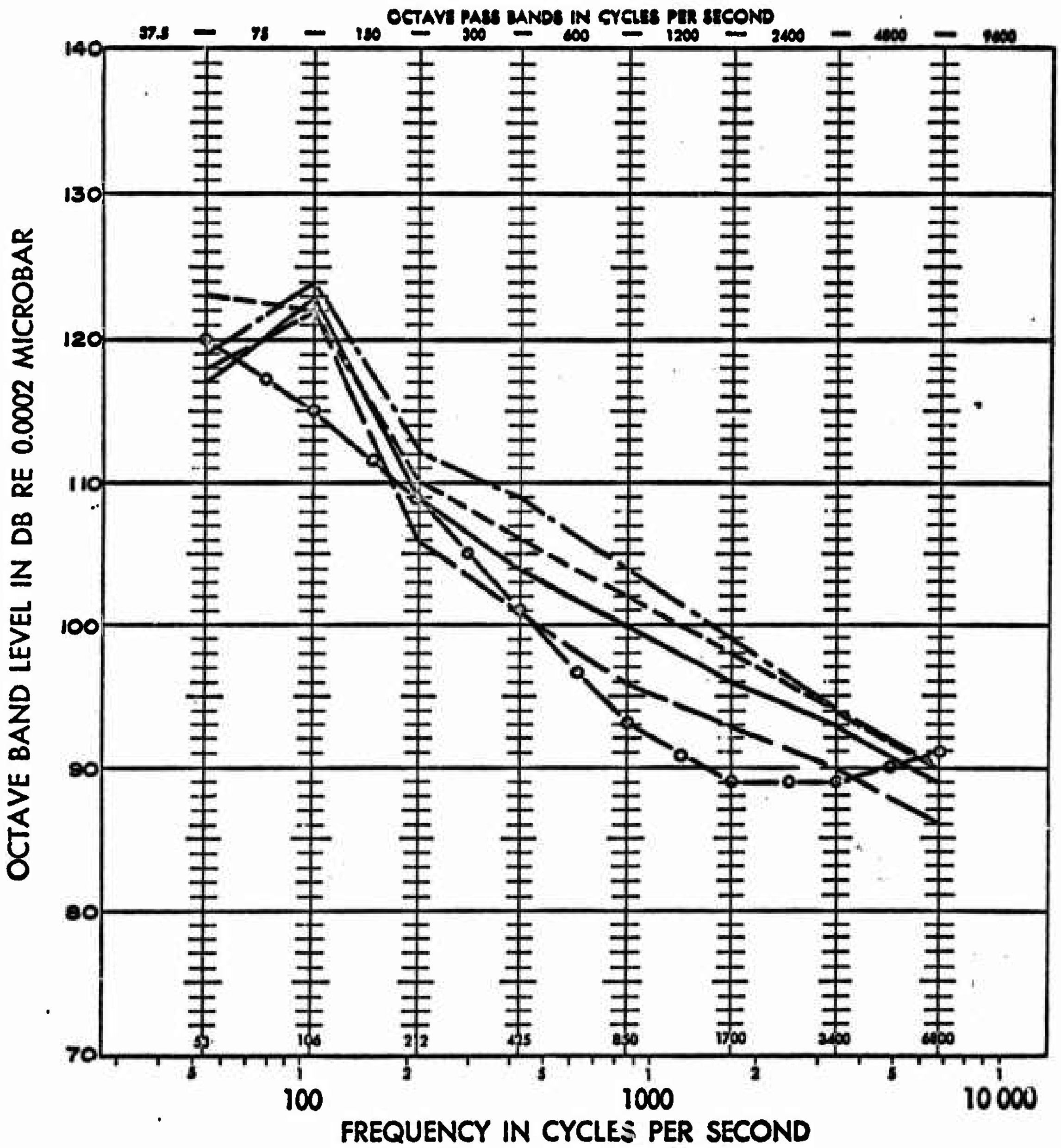


FIGURE 10. INTERNAL SOUND SPECTRA AT THE DRIVER'S POSITION.



10 MPH, HATCHES OPEN
10 MPH, HATCHES CLOSED
20 MPH, HATCHES OPEN
20 MPH, HATCHES CLOSED
HEL STANDARD S-1-63

FIGURE 11. INTERNAL SOUND SPECTRA AT THE PASSENGER'S POSITION IN THE RIGHT REAR CORNER.

The following table illustrates interior sound level as a function of several parameters including speed and terrain. The terrains used for these comparative studies were a hard paved surface, a soft dry dirt surface, a 30° gravel hillside, and a small lake. The tests were made with all hatches closed.

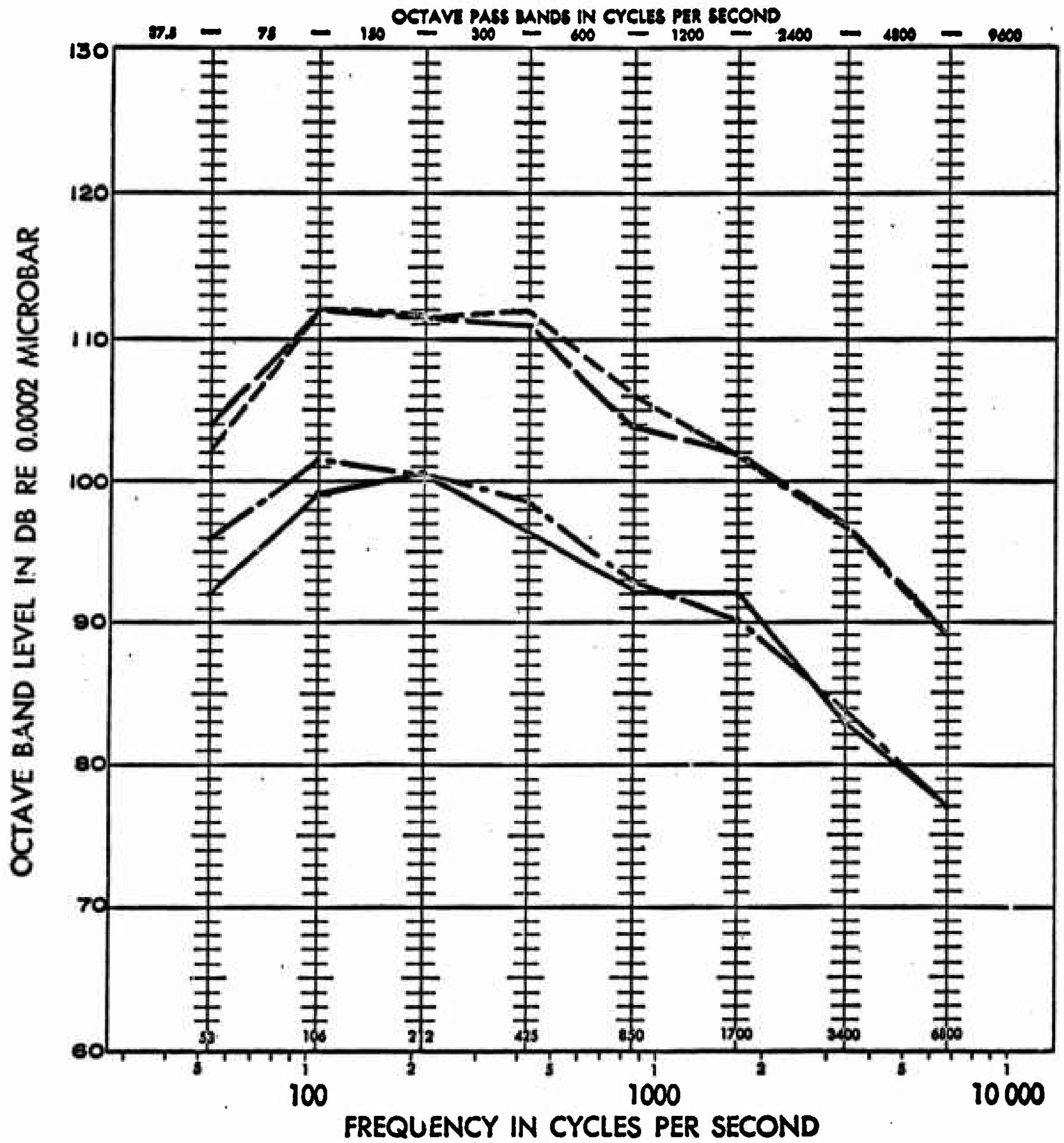
TABLE II. Variation of Internal Noise With Operating Conditions

Indicated Speed MPH	SPL, db Hard Track	Soft Track	Up Hill	Down Hill	In Water
5	106	105.5	106	105.5	105
10	116	115.5	116.5	114	113
15	119.5	119	118	118.5	115
20	120	120			
25	121	120			
30	123	122.5			

These tests show that increasing the vehicle speed from 5 mph to 30 mph increased the internal sound level at the center of the passenger comparment by 17 db. The greatest increase occurred between 5 mph and 10 mph.

Driving on a hard surface instead of a soft one increased the internal sound level slightly at all speeds. The increase, however, was less than 1 db. Octave band comparisons of operating on a hard surface to a soft surface is given in Figure 12. Running up and down a 30° hill had little effect on the overall level. Swimming the vehicle showed a reduction ranging from 1 to 4 db at the indicated speeds at which measurements were made.

It may be noted that of all the parameters tested, vehicle speed had the most important effect upon internal noise levels.



5 MPH, HARD PAVED ROAD
25 MPH, HARD PAVED ROAD
5 MPH, SOFT DIRT ROAD
25 MPH, SOFT DIRT ROAD

FIGURE 12. INTERNAL SOUND SPECTRA SHOWING ROAD SURFACE EFFECTS.

It was noted that a considerable fluctuation (about 5 db) in noise level was apparent when running at a desired speed. The policy developed was to record the average values only. The results were repeatable to within 1 db throughout the testing period. It should be noted that these variations in noise levels are actual modulations in the levels and not caused by the equipment.

4. External Noise Studies

Also of importance from the detectability viewpoint is the external noise level and spectrum produced by the vehicle. Tests were conducted similar to the internal tests to evaluate the externally produced noise as a function of operating conditions. The external level produced by the engine under idling conditions measured at 50 feet from the vehicle ranged from 81 db at 750 rpm, 86 db at 1500 rpm, to 90 db at 2500 rpm. Again, it was found that these noise levels were not repeatable throughout the testing period, as differences of 5 db were noted in the level at 2500 rpm. However, outside noise measurements have the additional variables of wind speed and direction and ambient noise fluctuations to be considered. Octave band analyses of the external engine noise are given in Figure 13. The noise level at 2500 rpm was found to decrease by 3.7 db as the sound level meter was raised from ground level to a position 8 feet above the ground.

Driveby tests were used as a method of measuring external noise with the vehicle in motion. Again, measurements were made at a distance of 50 feet from the test track. The noise levels were recorded at their maximum values (i.e., when the vehicle was closest to the measuring position). The external noise spectra observed on the standard test course are given in Figure 14 for speeds of 10 mph and 20 mph.

Measurements were made varying speed and terrain to study their effects on external noise. Table III gives the external noise level as a function of these parameters. The test courses used were the same as those used for the internal noise measurements. For the hillside tests, measurements were made at a distance of only 25 feet away because of heavy brush farther away. However, the measured values were theoretically compensated so they are compatible with other test results.

Increasing the speed from 5 mph to 30 mph increased the external noise level by 15 to 16 db. Again, as in the case of the internal noise, the greatest change occurred between 5 mph and 10 mph.

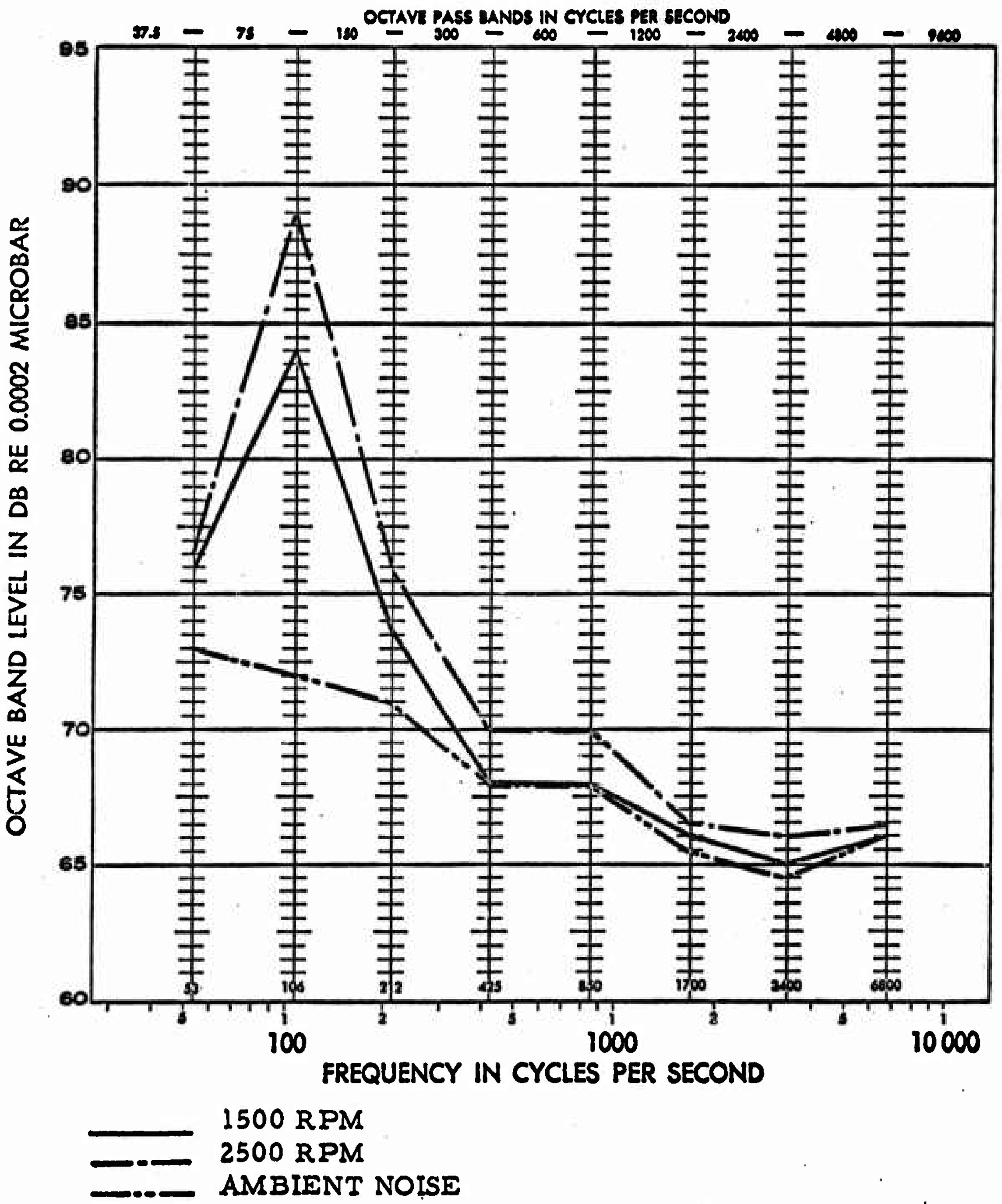


FIGURE 13. EXTERNAL SOUND SPECTRA WITH ENGINE IDLING, 50 FT FROM VEHICLE.

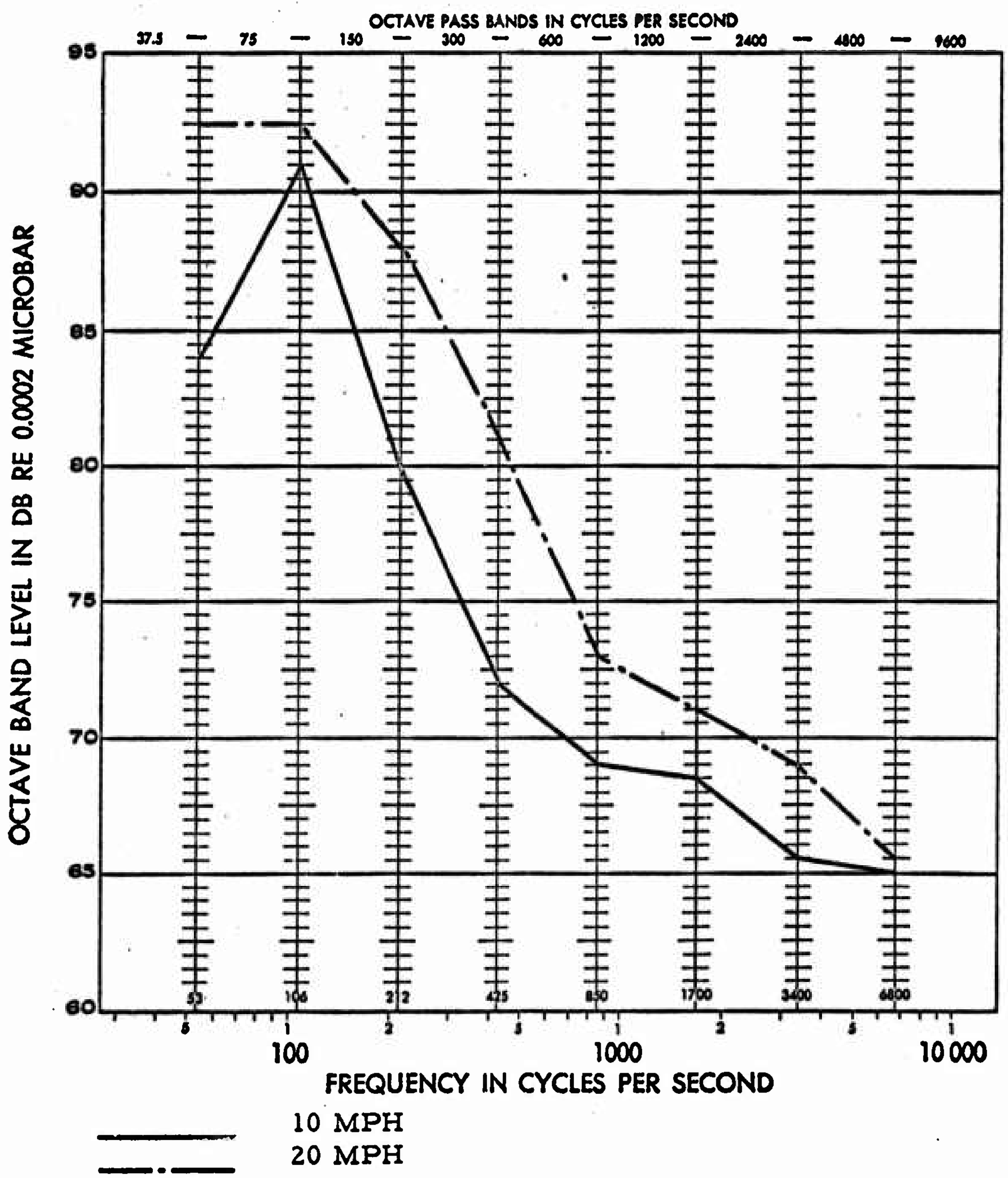


FIGURE 14. EXTERNAL SOUND SPECTRA, 50 FT FROM VEHICLE.

TABLE III. Variation of External Noise With Operating Conditions

Speed MPH	SPL, db Hard Track	Soft Track	Up Hill	Down Hill
5	85	82	87	86
10	94	91	93	93
15	97	96	95	96
20	98	96		
25	97	96		
30	100	98		

Driving by on a hard road produced 1 to 3 db more external noise than similar tests on a soft road. Octave band analyses comparing the hard track to the soft track are given in Figure 15. Driving by uphill and downhill produced about the same noise levels as on a level track, except at low speed it was slightly noisier on the hillside track.

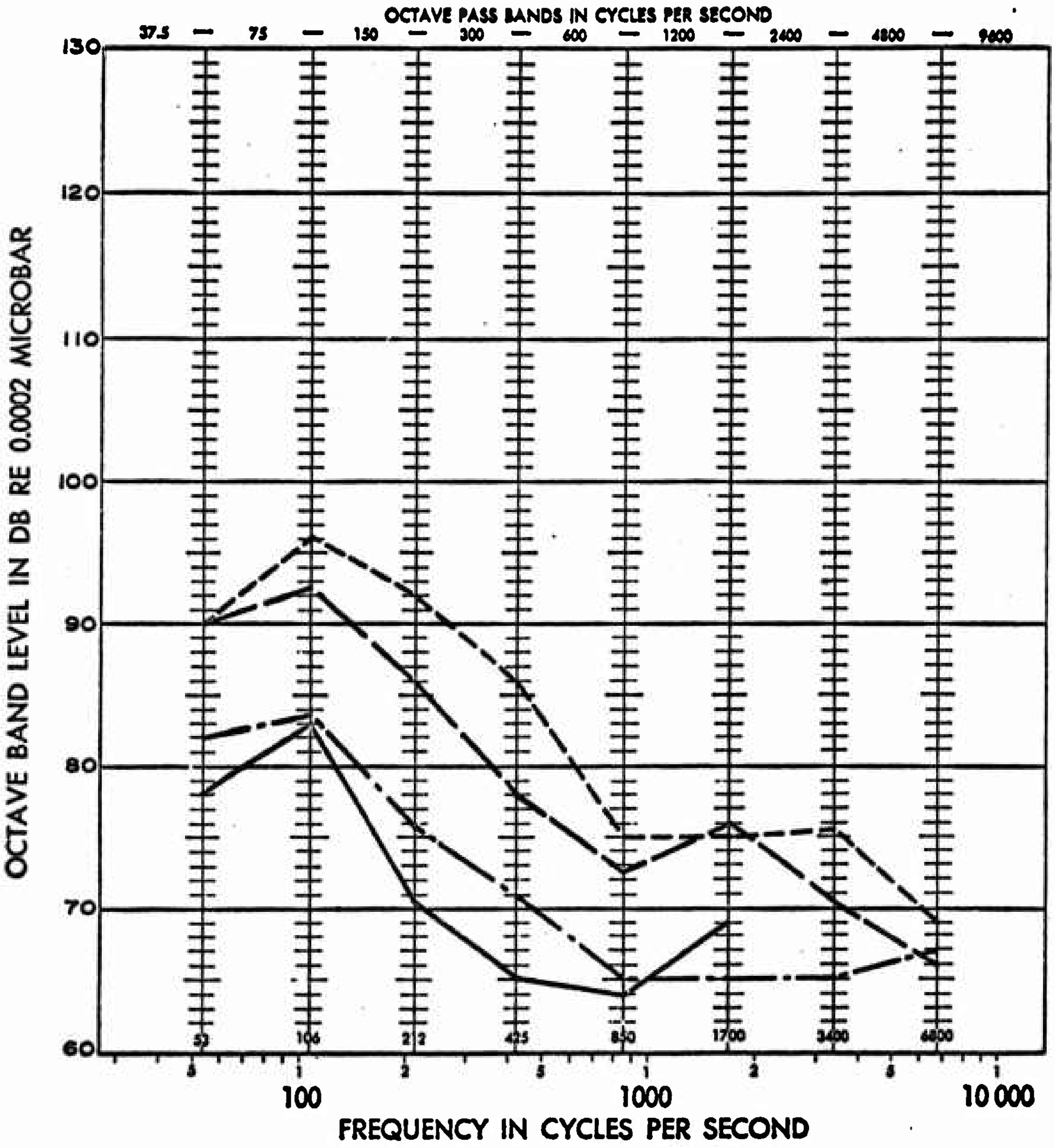
The testing that has been discussed thus far in this report summarize the noise levels that are created by the M-113 tracked vehicle as a function of practical operating conditions; thereby laying the foundation for further testing to evaluate the noise and vibration sources.

C. Evaluation of Vibration Characteristics

1. Vibration Criteria

In addition to mechanical vibrations creating noise which can prove detrimental to personnel and personnel performance, direct contact with vibrating elements can also cause effects of a serious or, at least, annoying nature. It is important that the efficiency and physical well being of occupants of military vehicles have not been compromised because of excessive vibration to the extent that they cannot perform effectively.

The knowledge of the effects of vibration on man is still in somewhat of an elementary state. Some criteria have been established and published, but these are limited in scope and lack clarity in meaning. The vibration criteria given in Figure 16 were taken from the General Radio



5 MPH, SOFT DIRT ROAD
25 MPH, SOFT DIRT ROAD
5 MPH, HARD PAVED ROAD
25 MPH, HARD PAVED ROAD

FIGURE 15. EXTERNAL SOUND SPECTRA SHOWING ROAD SURFACE EFFECT, 50 FT FROM VEHICLE.

Handbook of Noise Measurement[6] which summarizes the various published criteria to date. The three curves prepared by one of the investigators represent the threshold of perception, the threshold of discomfort, and the threshold of tolerance. These curves are the consolidated results of the work of a large number of experimenters. The personnel used for the testing were either standing, sitting, or lying on a support that was vibrated vertically or horizontally. There was some question of the exposure time used by many of the investigators. The dashed curve given in Figure 16 represents the work done by another investigator on the vertical vibration limits for automobile passenger comfort. This investigator considers only the effects of vibrations in the vertical direction on personnel sitting or standing on a hard seat. His formulas derived for passenger comfort as a function of frequency are presented at the top of the figure.

Brief measurements were made of the vibrations experienced by passengers in the M-113 tracked vehicle. Vibration transducers were mounted on the floor panel between the feet of passengers sitting on the right bench. Additional transducers were mounted on the right bench back rest on the opposite side of the passengers at the centers of their backs. Because the seats have cushions, no transducers were mounted on them. The exposure test consisted of accelerating from a standstill to a speed of about 30 mph on the level dirt track. Continuous records of the vibrations of the floor panel and back rest were made with the CEC oscillograph. The recorded vibration levels were quite intense reaching the maximum values indicated in Figure 16. These peak values were recorded at the individual resonances of the two panels. Even under non-resonant conditions the levels are still high. The criteria presented in Figure 16 are representative of the vibration effects on personnel that are supported by the vibrating element; however, in our tests, the vehicle floor panel and back rest only partially support the seated passenger. Thus, if the vibration of either of these elements becomes too strong, the natural tendency will be to break the contact with the surface or, at least, lessen the force exerted on the surface.

In summary, it is difficult to compare the vibrations experienced by occupants of the tracked vehicle to meaningful vibration criteria because of the limited nature of the published criteria and the complexity of the vibrations experienced in the vehicle. It is felt that although the vibration may not be of such a nature to inflict immediate damage to personnel, certainly a deteriorating effect must be produced on the performance of the individuals.

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6 Handbook of Noise Measurement, 5th Edition, Chapter 14, General Radio Company, West Concord, Massachusetts, 1963

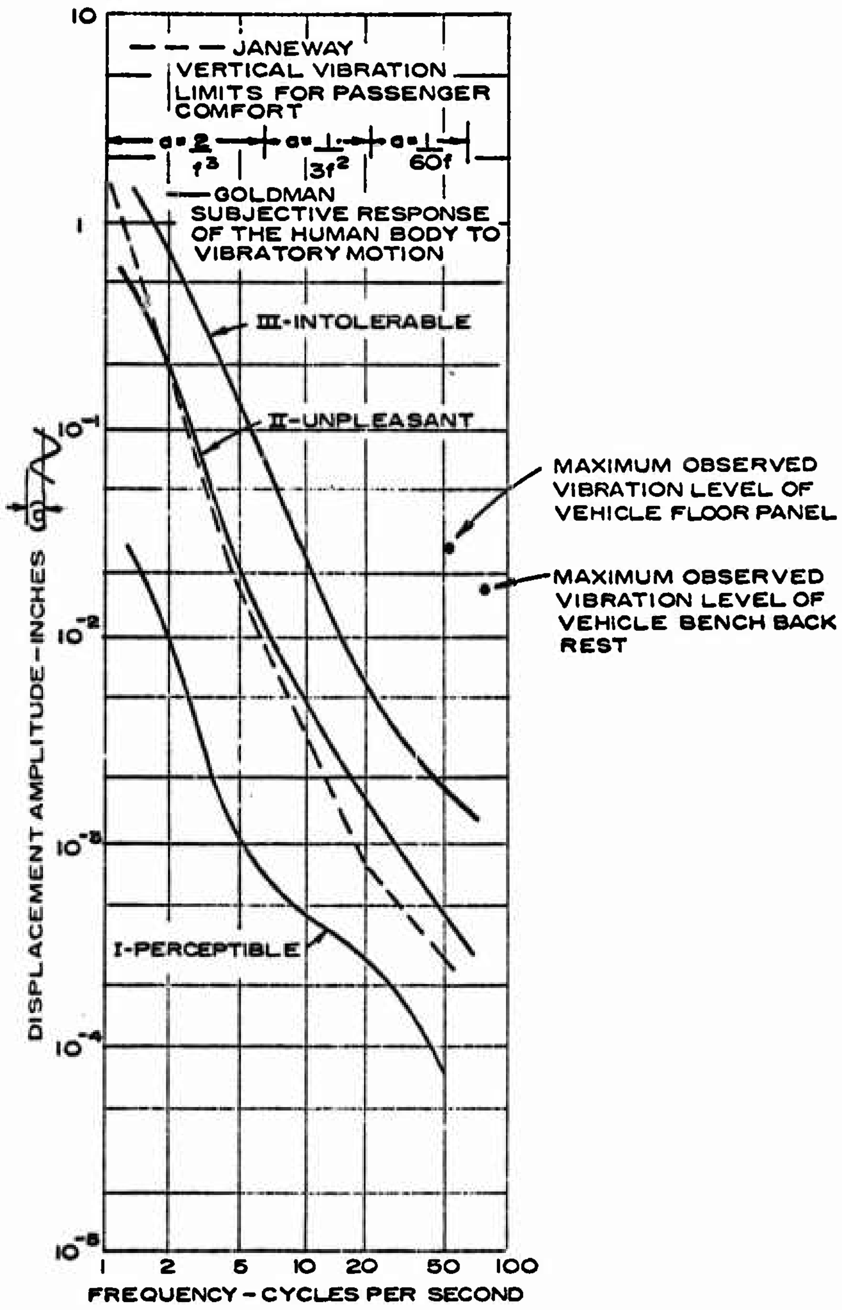


FIGURE 16. SAMPLE VIBRATION CRITERIA AND MAXIMUM VIBRATION LEVELS EXPERIENCED BY PASSENGERS SITTING ON THE RIGHT BENCH.

2. Theory of Noise Radiation From A Panel

In order to predict the sound intensity radiated by a vibrating panel on the basis of panel vibration measurements, it is necessary to know the vibrational characteristics of the element and its surface area. For an ideal case in which the panel acts as a piston in an infinite baffle and the surrounding air is not confined (ie, non-reverberant), the velocity of panel vibration is equal to particle velocity in the adjacent air (u), and the resulting pressure (p) is given by

$$p = Zu$$

For the semi infinite air medium, impedance Z is the product of density (ρ) and acoustic velocity (c). The radiated sound power (P_{WL})is the product p u, and radiating area. Thus,

$$P_{WL} = u^2 \rho c A$$

When the environment is reverberant, the acoustic impedance is no longer ρc, and reactive terms vary the relationship of p and u; however, power is the in-phase product of p and u and will not vary significantly. Thus, power is proportional to u^2A, ie

$$P_{WL} = K u^2 A \qquad \text{where K is a constant}$$

When multiple modes of panel resonance are experienced, it is apparent that the calculation of radiated power becomes much more complex. However, previous experience has shown that for such panels, a rather effective comparison of panel radiated noise in a given environment can be made by averaging the vibration velocity across the panel, and thus the product u^2A becomes a rather convenient index for comparing the noise radiating potential for such structural panels.

3. Preliminary Vibration Measurements

In order to gain an understanding of the vibratory nature of the vehicle, several exploratory vibration surveys were made. These surveys were first conducted with various hand held instruments and transducers while the engine was idling and with the vehicle in motion. These initial tests were made for comparing response of the various vibrating elements, and for studying the mode shapes of selected panels in detail.

Table IV lists vibration acceleration magnitudes measured at various points in the vehicle with the engine idling at 2500 rpm. The results

TABLE IV. VIBRATION SURVEY: ENGINE IDLING 2500 RPM

Surface	Acceleration Rms, cm/sec^2
Engine access panel beside driver	925
Engine access panel facing passenger compartment	873
Engine enclosure below shift lever	694
Front upper engine enclosure	551
Access plate on floor	490
Right armor floor under driver	436
Floor below center seat	390
Top metal engine enclosure	390
Floor by gas pedal	276
Left armor floor under driver	276
Radio shelf on left side	276
Right bench backrest	276
Floor panels (near front)	220
Instrument Panel	220
Top of left fender armor	196
Front sloping armor	196
Top of Battery Box	196
Left bench backrest	196
Floor panels (center area)	175
Left fender side armor	124
Top armor (Rt. rear corner)	124
Right armor wall	110
Central seat post	98
Left armor wall	98
Top armor	98
Access door to fuel tank	98
Ramp	87

given are the maximum measured values for the surfaces indicated. While the data are in terms of acceleration, it is possible to predict approximate generated sound pressure level from these results if both the amplitude and frequency distribution are known. As an example, the average measured accelerations of the large engine access panel in the passenger compartment at idling speeds of 750 rpm and 2500 rpm were found to be 190 cm/sec^2 and 627 cm/sec^2, respectively. Octave band analyses of the vibration spectra of the panel indicated that the 75-150 cps band was predominant at both speeds. Thus, using the average frequency over this band and assuming a vibrating piston source, the near-field radiated sound pressure level can be approximated by the following equations:

$$\bar{v} = \bar{a}/2\pi\bar{f}, \quad \bar{p} = \rho c \bar{v}, \quad SPL = 20 \log \frac{\bar{p}}{2 \times 10^{4}}$$

where

$\bar{a}$ = average measured acceleration, cm/sec^2

$\bar{f}$ = average measured frequency, cycles/sec

$\bar{v}$ = average calculated velocity, cm/sec

ρ = density of air, 1.28×10^{-3} gm/cm^3

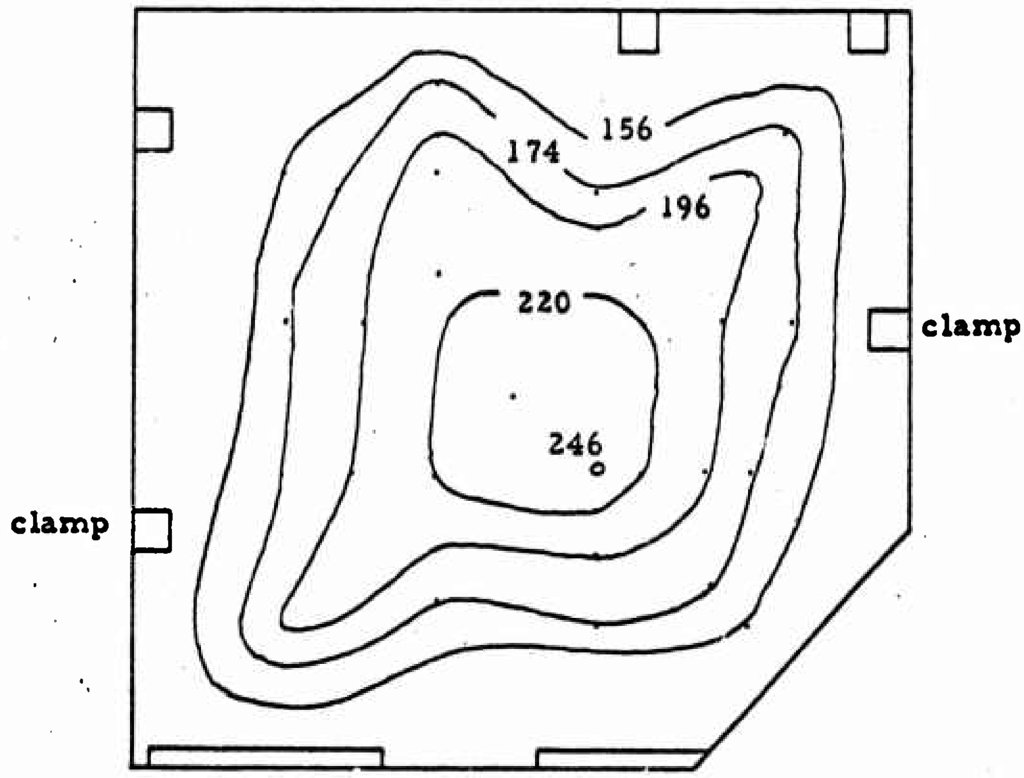
c = velocity of sound in air, 3.44×10^{4} cm/sec

$\bar{p}$ = average radiated sound pressure, dyne/cm^2

SPL= sound pressure level, db

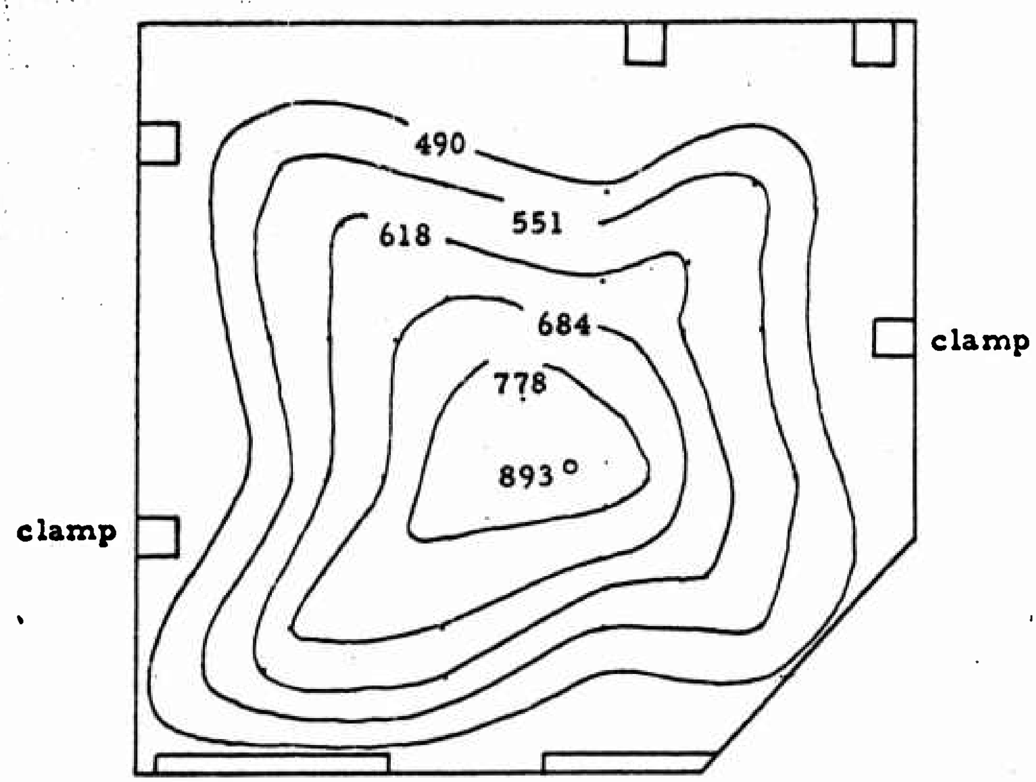
The computed sound pressure levels at the surface of the panel are 95.5 db at 750 rpm and 106 db at 2500 rpm. These values agree with the measured near-field sound pressure levels given in Figures 4 and 5. Thus, from these results and Table IV, it is apparent that the engine access panels are major noise radiators under idle conditions.

Vibration contour plots of the two engine access panels are given in Figures 17 and 18. These patterns indicate driven non-resonant vibration modes controlled by the clamps at the edges of the panels. The total vibration level at the center of the engine access panel in the passenger compartment was measured as a function of engine speed in order to determine resonant conditions. This is illustrated in Figure 19.



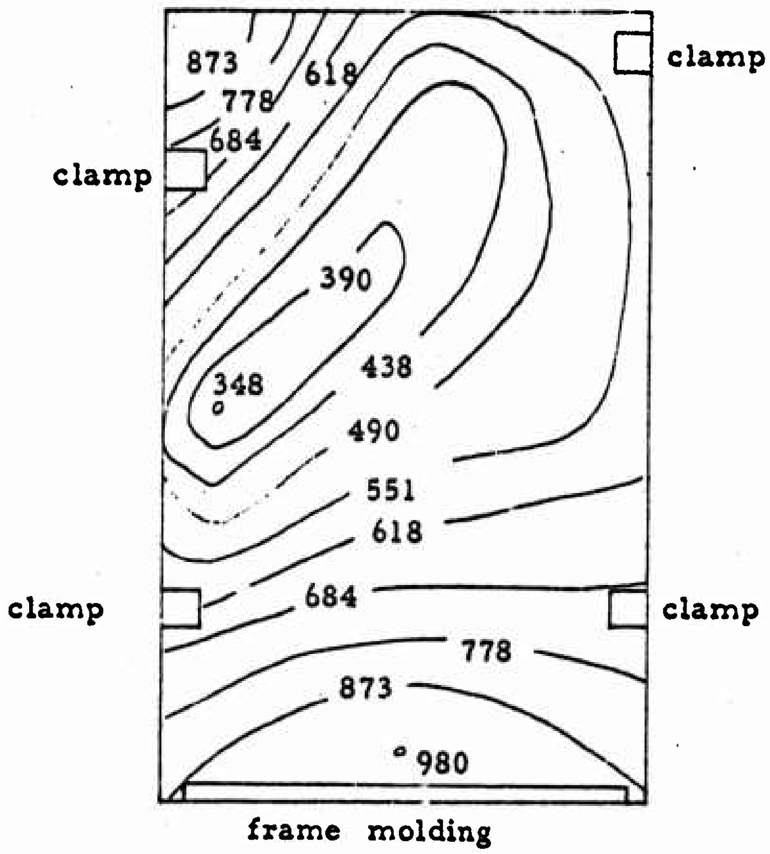
A. Engine Idling at 750 rpm

(Contours are acceleration in cm/sec^2 rms)



B. Engine Idling at 2500 rpm

FIGURE 17 - VIBRATION PATTERNS FOR ENGINE ACCESS PANEL.



(Contours are acceleration in cm/sec^2 , rms)
Engine Idling at 2500 rpm

FIGURE 18. VIBRATION PATTERN, DRIVER'S PANEL.

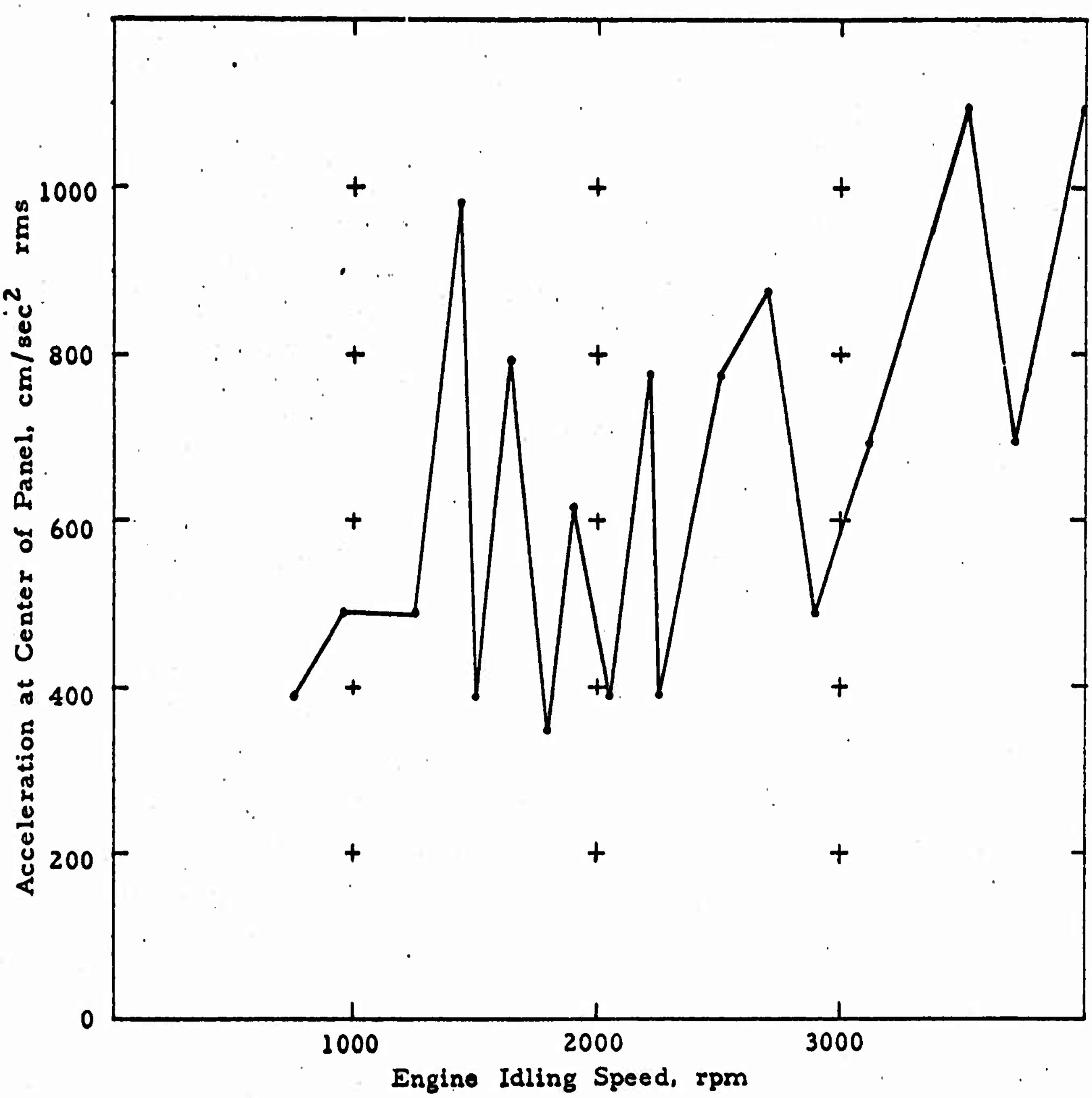


FIGURE 19. VIBRATION RESONANCES OF LARGE ACCESS PANEL.

4. Vibration Survey for Potential Noise Radiator Evaluation

a. Testing Procedure

With the instrumentation system described in Section II-A, a thorough vibration survey was conducted operating the vehicle through a variety of typical driving tests. There were several objectives in this survey, including identification of the more serious noise radiators in the vehicle and possible assessment of the vibration sources and transmission paths between the sources and the noise radiators.

Because of the complexity and the impracticality of analyzing the vibratory mode shape as a function of operating condition for all the panels in the vehicle with their different shapes and mountings, a statistical method was developed for measuring vibrational velocity and, thereby, calculating the radiated sound. The measuring technique used was to record the vibration at several points on each panel, thus obtaining representation averages of the vibrational velocity, regardless of the mode of vibration or mixture of modes. One transducer was always mounted near the center of each panel or point of least stiffness, in order to measure vibrations of the simplest primary mode. Other transducers were placed 1/3, 1/2, and 2/3 of the way toward various corners and edges, to sample some of the higher modes of vibration. For possible correlation of panel vibration to radiated noise, a microphone was mounted about 6" away from the panel under test, and sound was recorded along with vibration throughout the testing sequence. Samples of the recorded data are given in Figures 20-24.

The following test procedure was used for exploring the vibrations of the vehicle:

(1) Gear Range 3-6, speed 10 mph on level pavement

(2) Same, speed 10 mph up 15° paved hill

(3) Same, 90° left and right turns, 10 mph on level pavement

(4) Same, accelerating to top speed (25-35 mph) down 30° dirt hill

(5) Same, up 30° dirt hill, top speed

(6) Same, 5 mph down 30° gravel hill

(7) Range 1-2, accelerate to 10 mph, stop. Level dirt road

(8) Range 3-6, accelerate to 30 mph, stop. Level dirt road

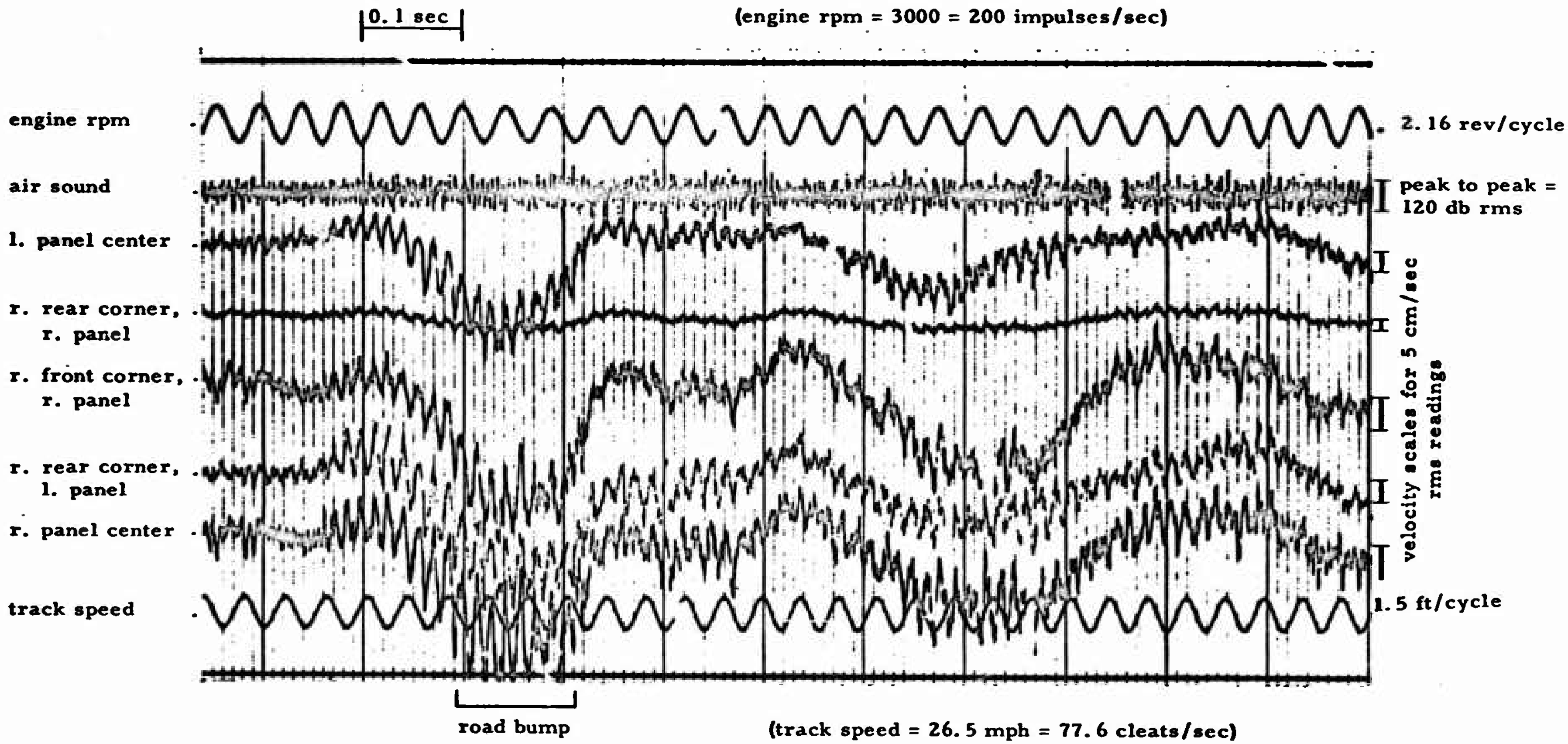


FIGURE 20. VIBRATION RECORD, REAR SUB-FLOOR PANELS, DOWNHILL ON DIRT

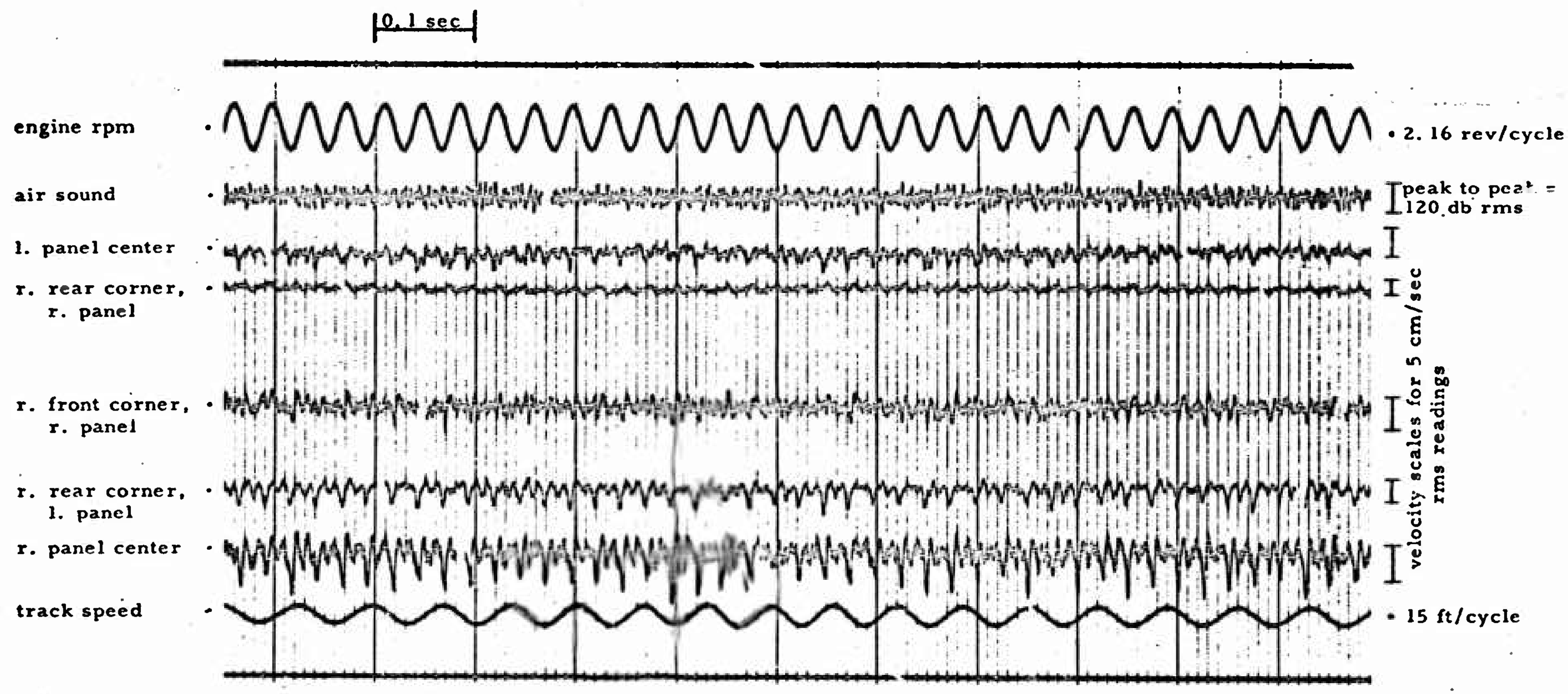


FIGURE 21. VIBRATION RECORD, REAR SUB-FLOOR PANELS, SWIMMING

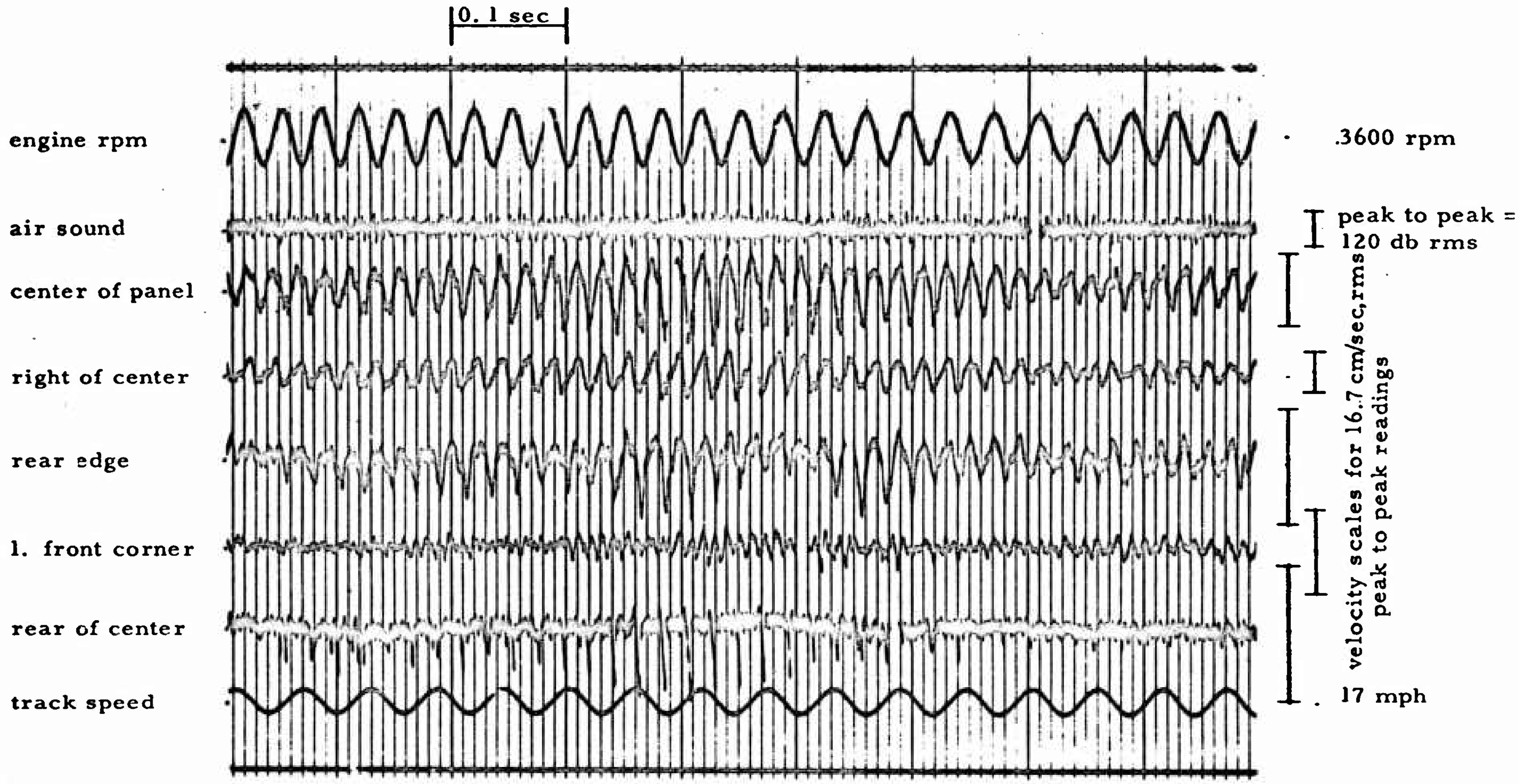
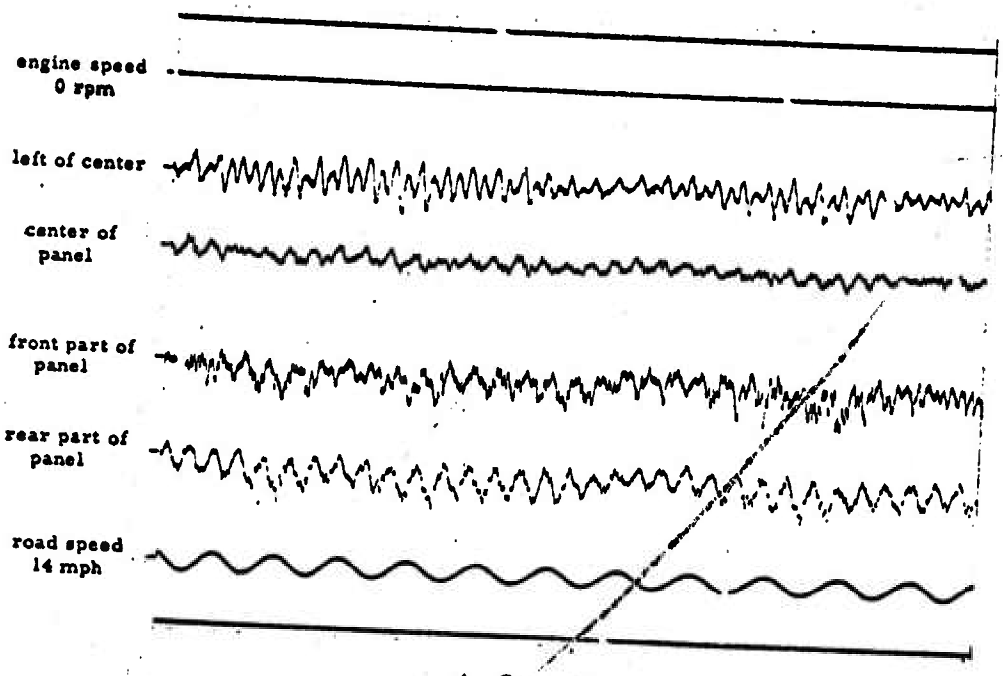
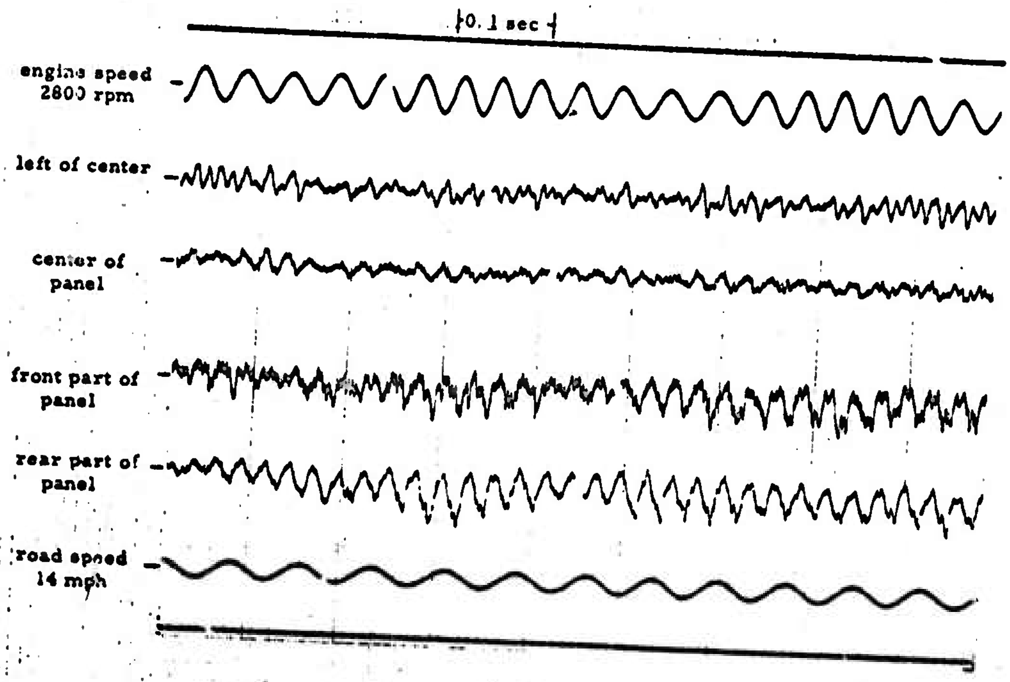


FIGURE 22. VIBRATION RECORD, RIGHT VEHICLE FLOOR PANEL, LEVEL DIRT ILLUSTRATING RESONANCE CONDITION



A. Coasting Downhill



B. Driving Uphill

FIGURE 23. COMPARISON OF VIBRATION OF RIGHT FRONT FLOORING PANEL COASTING DOWNHILL TO DRIVING UPHILL

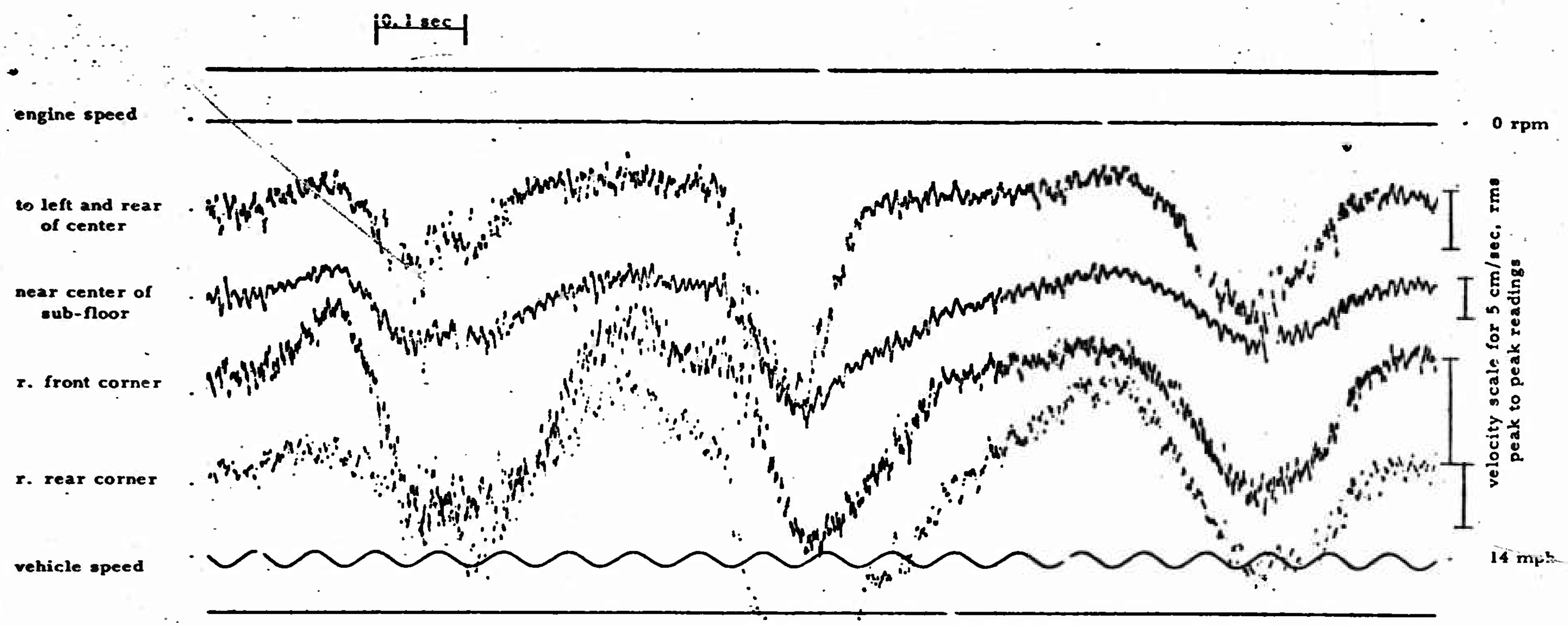


FIGURE 24. VIBRATION RECORD, VEHICLE SUB-FLOOR, SHOWING EFFECT OF COASTING OVER SEVERE ROAD BUMP

(9) Reverse, accelerate to 4000 rpm, stop. Level dirt road

(10) Range 3-4, accelerate to indicated 15 mph, Swimming in lake.

(11) Same, full left and right turns, 15 mph, swimming

(12) Reverse, accelerate to 4000 rpm, swimming

(13) Coasting with engine off down 30° gravel hill

(14) Coasting over sharp 6" bump at bottom of hill, about 15 mph

(15) Driving up 30° gravel hill, duplicating speed spectrum coasting down hill

b. Comparison of Noise Radiators

The method used to analyze the data was to measure the peak-to-peak amplitude of the vibrational velocity envelope in its strongest portion for each transducer channel for each test. The measurements were converted to rms velocity in cm/sec by means of the calibration data recorded on the same day that these tests were made. In the visual analysis process, it was frequently not possible to correlate each major frequency component with its proper amplitude; therefore, the policy developed was to measure only the overall amplitude and to list the predominant frequencies without reference to individual significance. On the other hand, when strong resonances occurred, each dominant frequency was tabulated with its own maximum amplitude. An example of the data sheets compiled is shown in Figure 25.

The dominant frequencies for each panel that were noted during the testing procedure were then plotted vs envelope or resonance amplitude. Some examples of the resulting "pseudo-spectra" are shown in Figures 26-30. The first three of these represent some of the noisiest sound-radiating surfaces discovered. These thin panels resonate at low frequencies (less than 100 cps) and high amplitudes, and the stiffer panels tend to resonate at higher frequencies and smaller amplitudes as shown by Figure 29. The effect of going from land to water in suppressing high frequencies can be seen from Figures 28 and 30.

Similar to the acoustical measurements, the vibration results indicate that speed has an important effect upon panel vibrational velocity, and that other parameters such as variations in terrain, climbing up and

V Series
04-1421

Date: 2-11-64
Name: [illegible]

	center, l. rear panel	r. rear corner, r. rear panel	r. front corner, r. rear panel	r. rear corner, l. rear panel	center, r. rear panel	air sound db
In Water [illegible] Test Rig 4	2.3	2	1.3	1.8	2.8	110
1500 RPM	40	40	100	50	40	200
15 MPH	120	120	150	120	150	—
	—	—	—	—	—	—
In Water – Left Turn	2.6	2	2.7	2.1	2.9	114
1750 RPM	55	25	30	60	28	210
10 MPH	120	80	150	100	120	—
	—	—	—	—	—	—
In Water – Right Turn	2.3	1.5	1.8	2.0	2.6	112
1900 RPM	90	90	90	70	70	200
12 MPH	—	—	—	—	—	
	—	—	—	—	—	
In Water – Reverse	1.8	1	1.7	1.5	1.8	112
2000 RPM	100	100	100	100	100	200
15 MPH	—					
	—	—	—	—	—	—
On Land – Up Paved Hill	3	2	2.5	2.8	2.5	117
2000 RPM	80	80	70	80	80	200
12 MPH	120	160	200	100	100	—
	—	—	—	—	—	—
On Land – Left Turn	2.5	1.8	2.7	2.8	2.3	115
1500 RPM	90	70	70	70	80	190
8 MPH	160	—	100	100	120	—
	—	—	—	—	—	—
On Land – Right Turn	1.[illegible]	1.2	2	1.7	2	116
2500 RPM	70	70	70	70	70	200
12 MPH	150	100	150	150	150	—
	—	—	—	—	—	—
On Land – Down Grade Fast	5	2	4	4.1	4.5	124
3400 RPM	70	70	80	70	100	300
30 MPH	200	200	200	250	2[illegible]	—
	—	—	—	—	—	—

top number: largest persisting rms velocity, cm/sec
lower numbers: most prominent frequencies, cps
(Series V refers to the test of two rear sub-floor panels)

FIGURE 25. EXAMPLE OF DATA SAMPLE FROM CEC RECORD

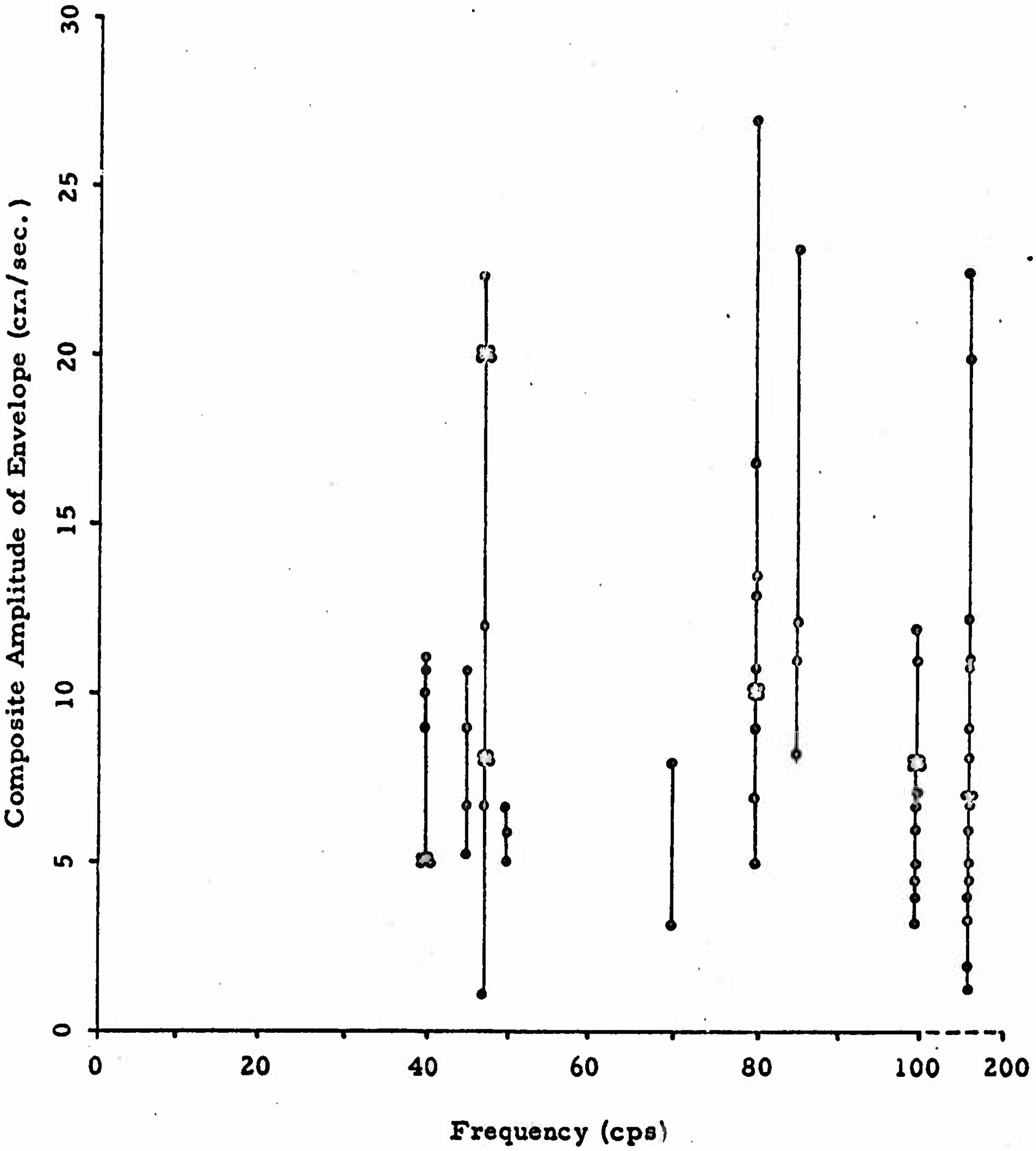


FIGURE 26. PSUEDO - SPECTRUM, RIGHT FRONT FLOORING PLATE, ON LAND.

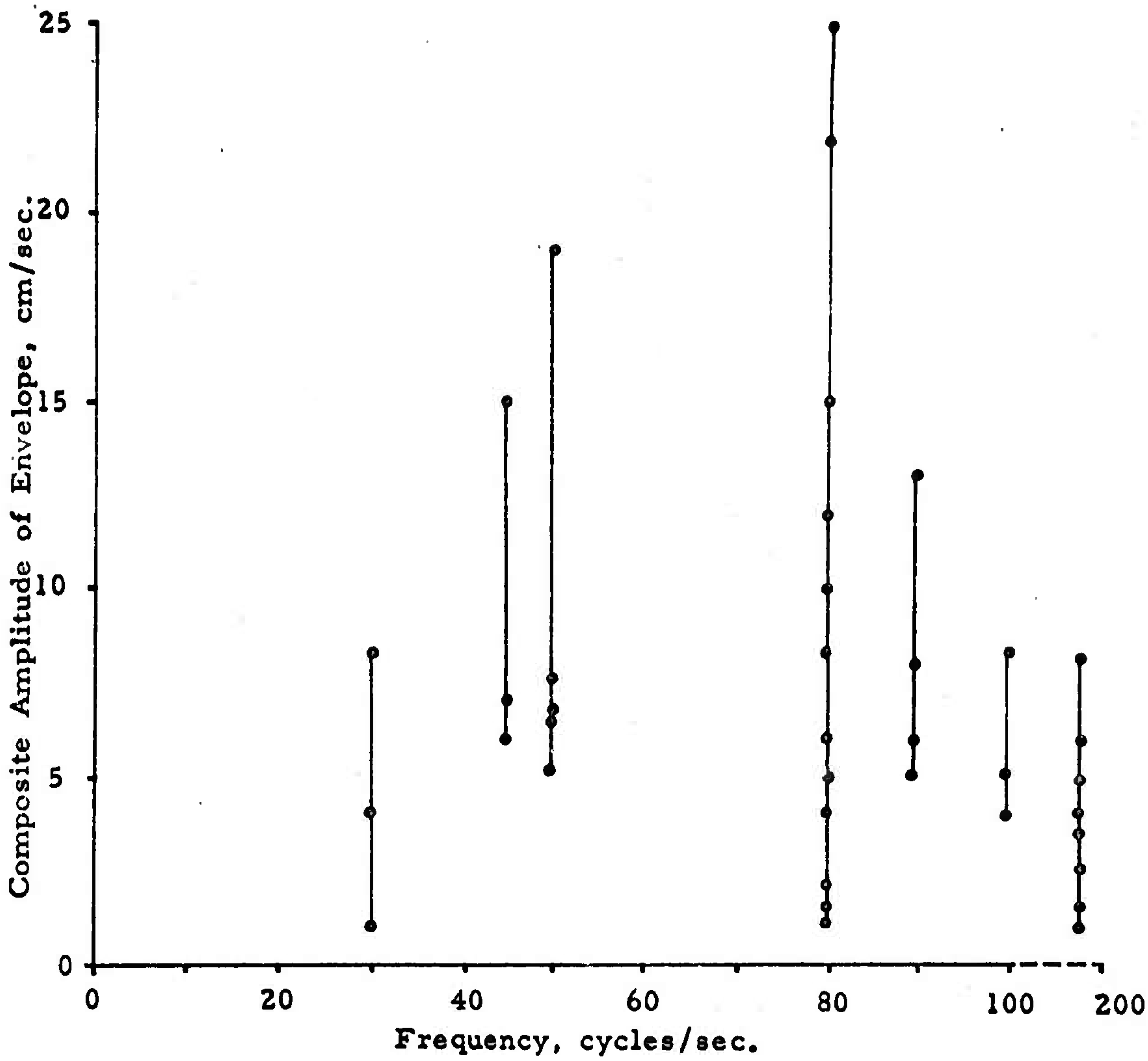


FIGURE 27. PSUEDO-SPECTRUM, FUEL TANK WALL (TANK 1/3 FULL), ON LAND.

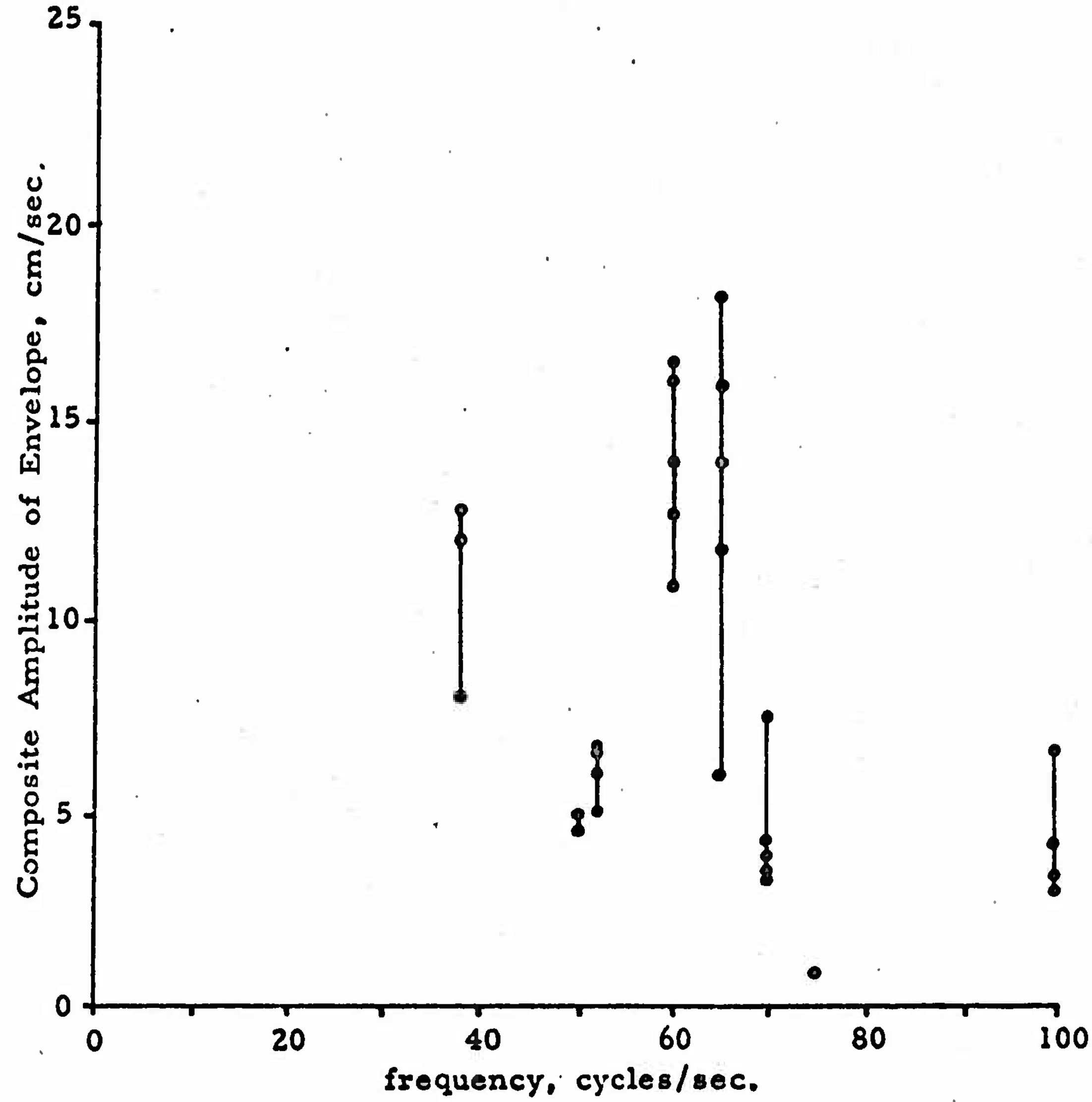


FIGURE 28. PSUEDO-SPECTRUM, DRIVER'S ENGINE ACCESS PANEL, ON LAND.

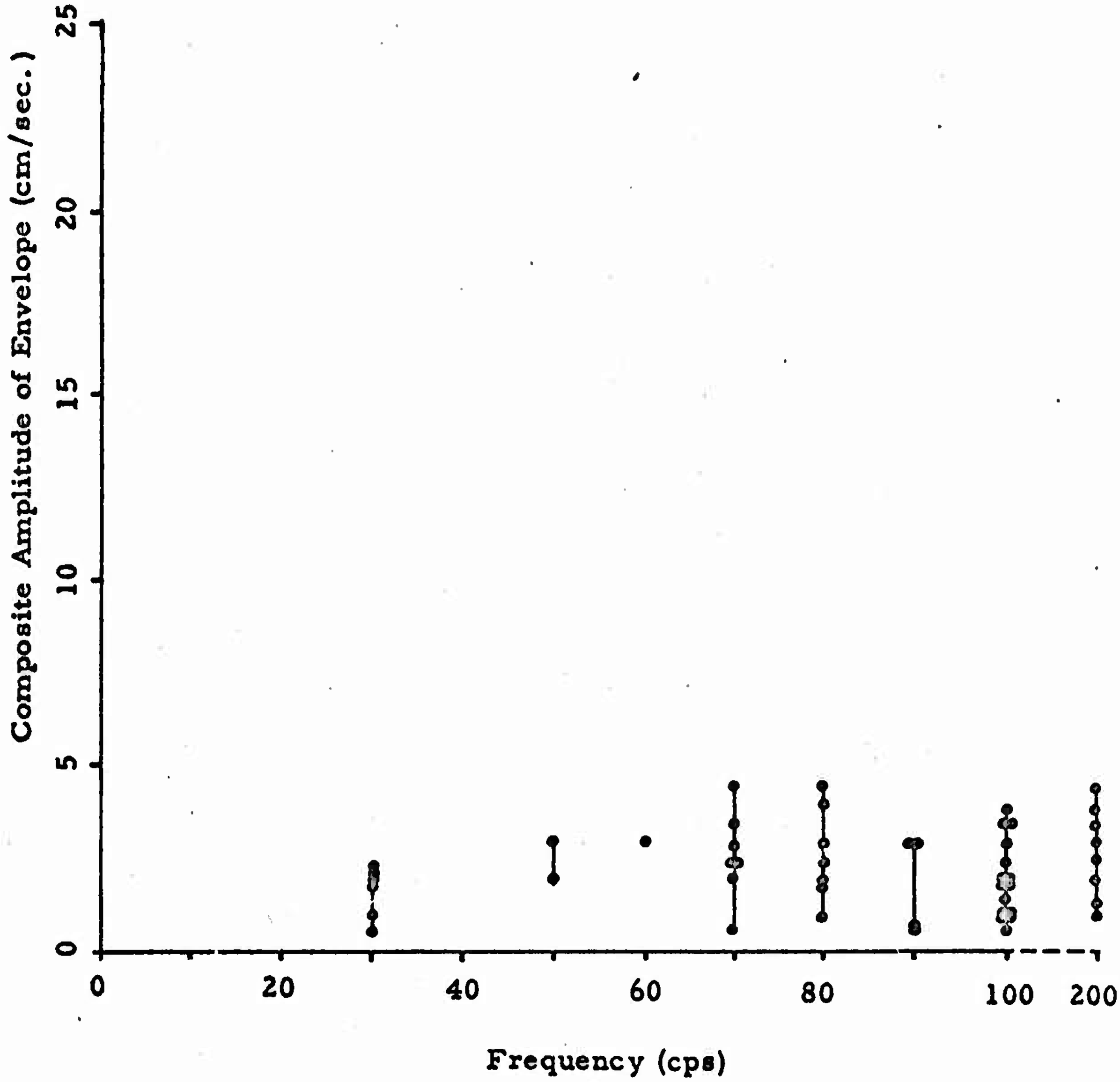


FIGURE 29. PSUEDO-SPECTRUM, LEFT ARMOR WALL, ON LAND.

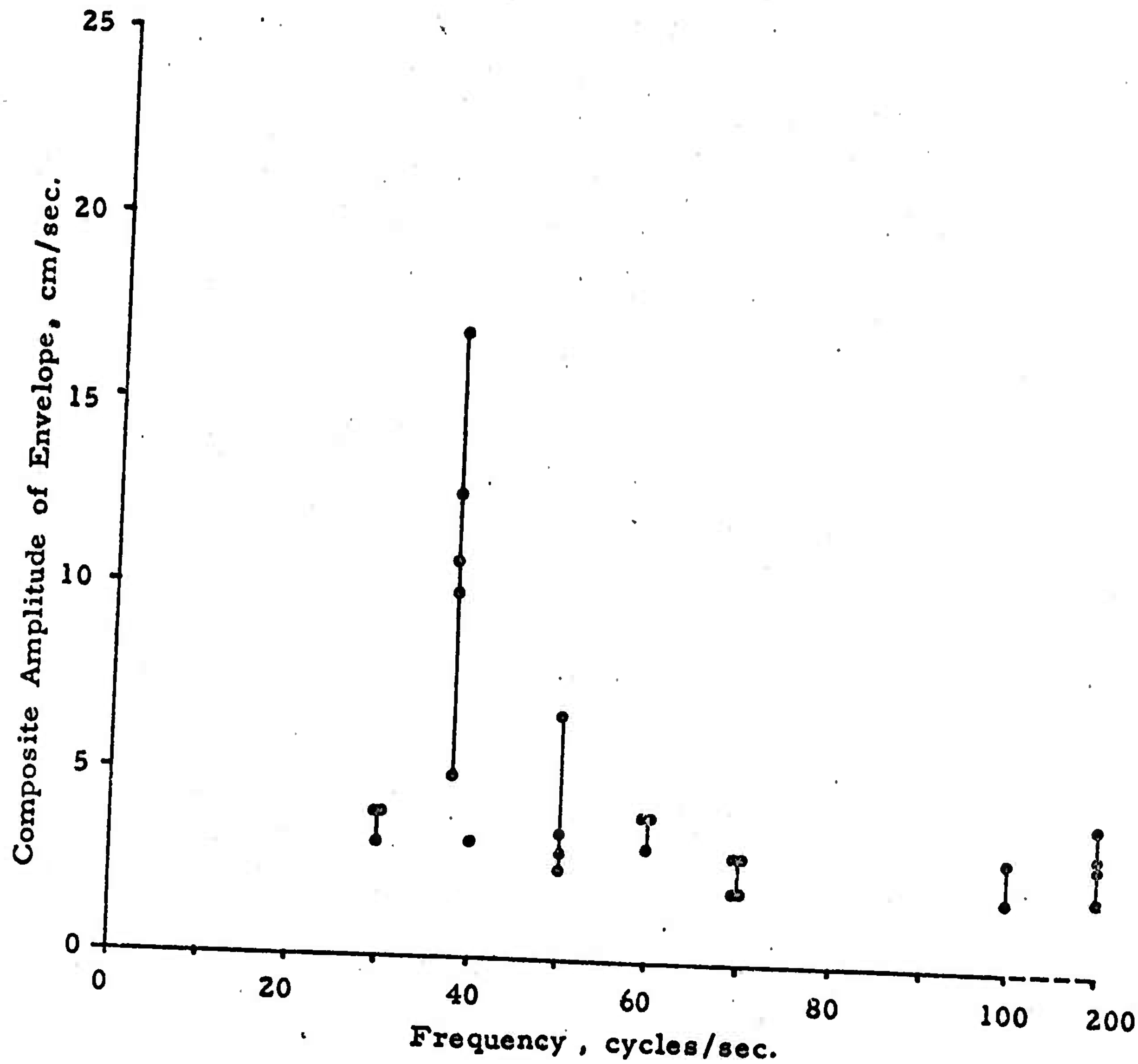


FIGURE 30. PSEUDO-SPECTRUM, DRIVER'S ENGINE-ACCESS PANEL, IN WATER.

down hills, turning, operating in water, and operating in different gear ranges, have a much lesser effect. Table V gives the velocities measured at each transducer on various panels at approximately 15 mph (12 mph to 16 mph) comparing operating on land to operating in water. It must be realized that comparing the vibration of several panels at one particular speed gives only a limited insight into the radiating importance of each panel. This is because each panel has different resonant frequencies, and since vibration response is a function of speed, the panels will resonate at different speeds. It was consistently noted throughout the survey that resonances would occur whenever the track-sprocket engagement frequency matched the various panel resonant frequencies. This track-sprocket engagement frequency in cps is approximately three times the vehicle speed in mph (ie, at 15 mph, the track-sprocket engagement frequency is about 45 cps). This frequency was also a major component under non-resonant conditions. Samples of panel velocities as a function of vehicle speed are given in Table VI for a thin floor panel and the stiffer sub-floor.

Tests were made coasting downhill with the engine off to study the engine and power transmission vibration contribution. Coasting data compared to driving uphill at a similar speed are tabulated in Table VII for various panels. The results indicate that the vibration induced by the engine has little effect on overall velocity amplitudes. This is also illustrated by Figure 23. Running over severe road bumps produces strong very low frequency (2-10 cps) velocity components in the vertical direction as illustrated by comparing Figures 20, 21 and 24. This results from excitation of the vehicle body-spring system at its natural frequency. There was also a noticeable "ringing" effect of higher frequencies at the peaks of the low frequency cycles.

In order to compare the noise-radiating importance of the various vehicle panels, the following evaluation technique was used. The test conditions which in general gave the strongest vibrations on land or in water were selected for making the panel radiation studies. The fast downhill run and the acceleration test in water were chosen as the noisiest conditions. Next, the maximum measured velocities for each panel were selected from these two tests, regardless of the speed. In order to arrive at a logical estimate of average velocity for the whole panel at its noisiest condition, a drawing was made showing the transducer locations. The panel area was then divided into "rings", similar in shape to the relatively stiff panel perimeter. The average velocity amplitude of each ring was taken to be the linear average of the average amplitudes recorded on its outer perimeter and its inner perimeter by the various transducers located on its edges (and center, for the center area). The fixed panel edges were assumed to vibrate with zero amplitude (resonant mode vibration). Examples are given in Figures 31-34. In each case, the area A, of each ring and its average rms velocity, u, were tabulated as shown, both on land and in water.

TABLE V. VIBRATION SURVEY AT 15 MPH*

Panel Description	Test Condition	Rms Vibration Velocity, cm/sec Transducer Number				
		No.1	No.2	No.3	No.4	No.5
Eng. access panel, dr. comp.	on land	12.7	7.5	8.0	12.7	11.0
	in water	17.0	5.0	12.7	11.0	10.0
Left armor wall	on land	1.5	1.0	1.2	1.2	1.2
	in water	1.0	1.4	1.4	1.0	1.5
Fuel tank wall	on land	6.5	5.0	8.0	13.0	5.0
	in water	6.0	4.8	5.3	10.0	3.5
Eng. access panel, pass.comp.	on land	3.0	7.0	6.0	6.7	6.7
	in water	4.5	6.7	6.7	8.0	9.0
Eng. wall, pass.comp.	on land	1.2	1.8	2.0	3.5	1.4
	in water	1.5	2.0	2.0	4.5	1.8
Right armor wall	on land	1.5	1.0	1.2	1.2	1.8
	in water	1.2	1.2	2.0	1.6	2.0
Top of right fender	on land	2.0	2.0	2.0		
	in water	2.5	2.7	3.0		
Side of right fender	on land	2.0	2.0	2.0		
	in water	2.0	1.5	2.1		
Right bench backrest	on land	15.0	20.0	22.0		
	in water	20.0	50.0	77.0		
Right front floor panel	on land	7.0	12.0	11.0	6.0	4.0
	in water	15.0	14.0	16.0	4.0	7.0
Left front floor panel	on land	6.0	6.0	7.0	8.0	12.0
	in water	4.5	16.0	30.0		7.3
Rear floor panel	on land	7.0	6.0	10.0	9.0	12.0
	in water	6.5	5.0	9.0	8.0	9.0
Sub-floor	on land	3.0	2.0	3.0	2.0	3.0
	in water	4.5	4.5	3.7	4.0	4.4
Top armor	on land	3.0	2.5	3.3	4.0	2.7
	in water	2.0	2.5	3.0	2.0	2.5
Rear ramp	on land	7.5	2.5	4.0	1.5	3.0
	in water	11.0	2.8	8.0	1.0	2.7

* Actual speeds ranged from 12 to 16 MPH.

TABLE VI. TOTAL RMS VIBRATIONAL VELOCITY LEVELS OF TYPICAL PANELS AS A FUNCTION OF VEHICLE SPEED

A. Right Front Floor Panel

Vehicle Speed MPH	Test Condition	Velocity cm/sec Transducer No. 1	2	3	4	5
5	on land	3.0	3.0	4.0	2.5	2.0
5	in water	3.0	3.0	5.0	2.0	2.0
10	on land	10.0	10.0	6.0	5.0	3.0
10	in water	3.5	5.0	4.0	3.0	2.0
15	on land	7.0	12.0	11.0	6.0	4.0
15	in water	15.0	14.0	16.0	4.0	7.0
20	on land	8.0	8.0	6.0	5.0	4.0
25	on land	5.0	5.0	5.0	4.0	3.0
30	on land	12.3	23.3	11.0	8.0	8.0

B. Sub-Floor

Vehicle Speed MPH	Test Condition	1	2	3	4	5
5	on land	1.3	1.0	1.3	1.3	1.3
5	in water	0.7	0.5	0.7	0.5	0.8
10	on land	2.8	1.5	1.7	2.0	2.0
10	in water	2.2	2.0	2.2	2.2	2.2
15	on land	3.0	2.0	3.0	2.0	3.0
15	in water	4.5	4.5	3.7	4.0	4.4
20	on land	1.6	2.0	2.0	2.0	2.7
25	on land	2.0	1.8	2.0	2.0	2.0
30	on land	4.5	3.5	3.8	4.0	4.5

TABLE VII. RESULTS OF COASTING TESTS

Panel Description	Test Description	rms vibration velocity, cm/sec transducers 1	2	3	4
engine access panel, driver's	coasting 14 mph	6.0	9.0	11.0*	11.0
	uphill 14 mph - 2200 rpm	6.0	7.5	7.0*	11.0
engine access panel, passenger's	coasting 13 mph	6.0	7.0	5.5	8.0*
	uphill 13 mph - 3000 rpm	7.5	7.0	5.5	9.5*
sub-floor	coasting 15 mph	3.3	3.0	2.8	3.5
	uphill 15 mph - 3200 rpm	3.3	3.0	2.7	3.0
right front flooring panel	coasting 13 mph	12.0	10.0*	13.0	18.0
	uphill 13 mph - 3100 rpm	12.0	12.0*	11.0	13.0
right front flooring panel**	coasting 13 mph	12.5	10.0*	10.0	11.0
	uphill 13 mph - 2500 rpm	10.0	10.0*	11.0	13.0
left armor wall	coasting 15 mph	3.0	1.5	1.8	2.5
	uphill 15 mph - 3300 rpm	3.0	1.5	2.0	2.3
left armor wall**	coasting 15 mph	2.0	1.2	1.8	1.5
	uphill 15 mph - 3000 rpm	1.8	1.0	2.0	1.5
fuel tank wall (1/5 full)	coasting 15 mph	11.0*	6.0	4.5	5.5
	uphill 15 mph - 3000 rpm	12.0*	6.0	4.5	7.0
fuel tank wall (full)	coasting 15 mph	6.0*	5.0	5.0	5.0
	uphill 15 mph - 2500 rpm	6.5*	5.0	5.0	5.0

TABLE VII. (Cont'd)

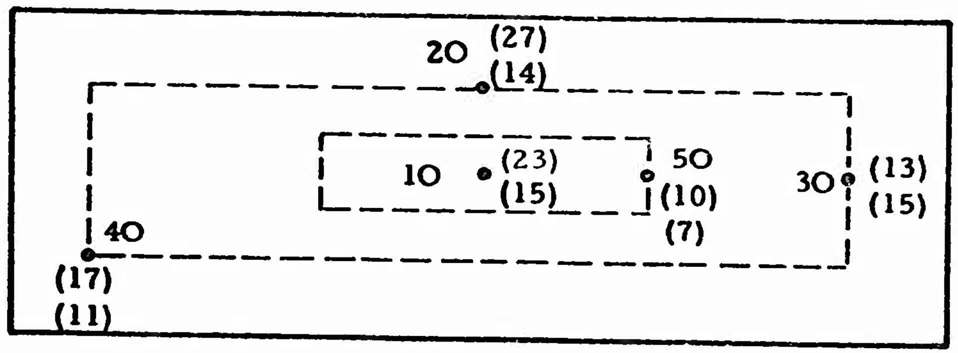
Panel Description	Test Description	rms vibration velocity, cm/sec transducers 1	2	3	4
battery box top(transducers 2&3) side(transducers 1&4)	coasting 13 mph	18.0	13.0	19.0*	17.0*
	uphill 13 mph - 3000 rpm	18.0	11.0	14.0*	16.0*
b nch back rests left(transducers 2&3) right(transducers 1&4)	coasting 16 mph	10.0	11.0*	15.0	16.0*
	uphill 16 mph - 3500 rpm	10.0	12.0*	13.0	13.0*
bench back rests**	coasting 16 mph	9.0	8.5*	11.5	10.0*
	uphill 16 mph - 3300 rpm	12.0	7.0*	9.5	13.0*
left bench back rest (with channel)	coasting 16 mph		6.0*	6.0	
	uphill 16 mph - 3500 rpm		6.5*	6.0	

* Transducer at center of panel

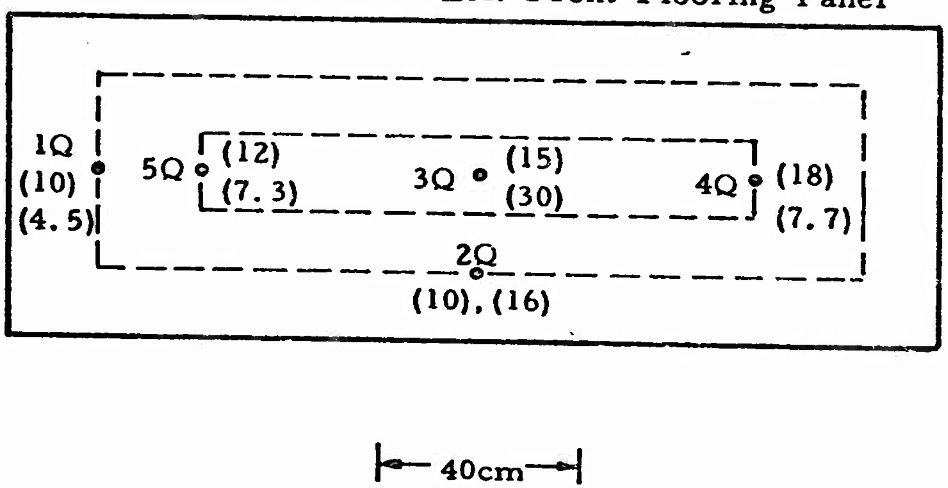
** Tests with rubber idlers on vehicle

Notation: Transducer numbering for this table is not necessarily the same as for previous tables since the testing was conducted at a different time.

Right Front Flooring Panel



Left Front Flooring Panel



Down Grade Fast

Ring	A	u	u^2A
1	1024	16.5	279,000
2	4520	16.6	1,268,000
3	6996	9.5	633,000
			2,180,000

Accel. in Water

Ring	A	u	u^2A
1	1024	11.	124,000
2	4520	11.8	624,000
3	6996	6.7	311,000
			1,059,000

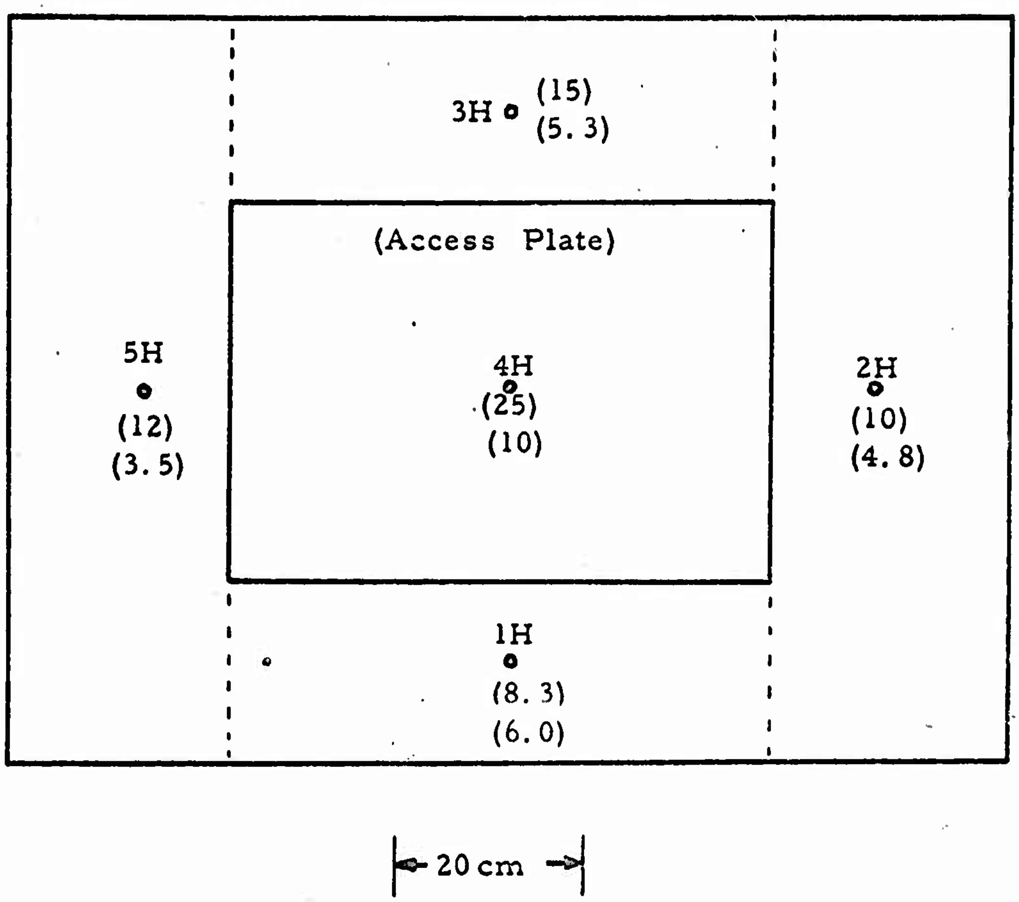
Down Grade Fast

Ring	A	u	u^2A
1	1792	15.	404,000
2	4608	12.5	719,000
3	6140	5.0	154,000
			1,277,000

Accel. in Water

Ring	A	u	u^2A
1	1792	15.	404,000
2	4608	8.6	342,000
3	6140	5.1	161,500
			907,500

FIGURE 31. ARRANGEMENT OF SAMPLING ON FLOORING PLATES.



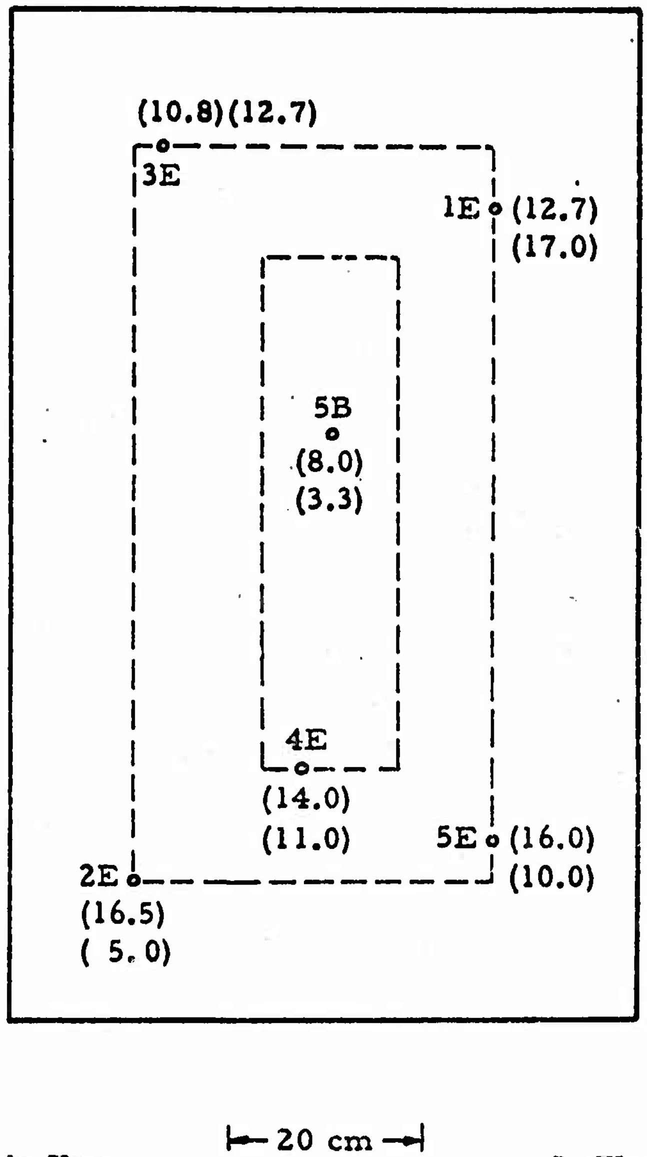
Down Grade Fast

Area No.	A	u	u^2A
1	2052	25.	1,282,000
2	1064	8.3	73,000
3	1748	12.	252,000
4	1064	15.	239,000
5	1748	10.	175,000
		Total	2,021,000

In Water 0 - 15 mph

Area No.	A	u	u^2A
1	2052	10.	205,000
2	1064	6.	38,300
3	1748	3.5	21,400
4	1064	5.3	29,900
5	1748	4.8	40,200
		Total	335,000

FIGURE 32. ARRANGEMENT OF SAMPLING ON FUEL TANK WALL.



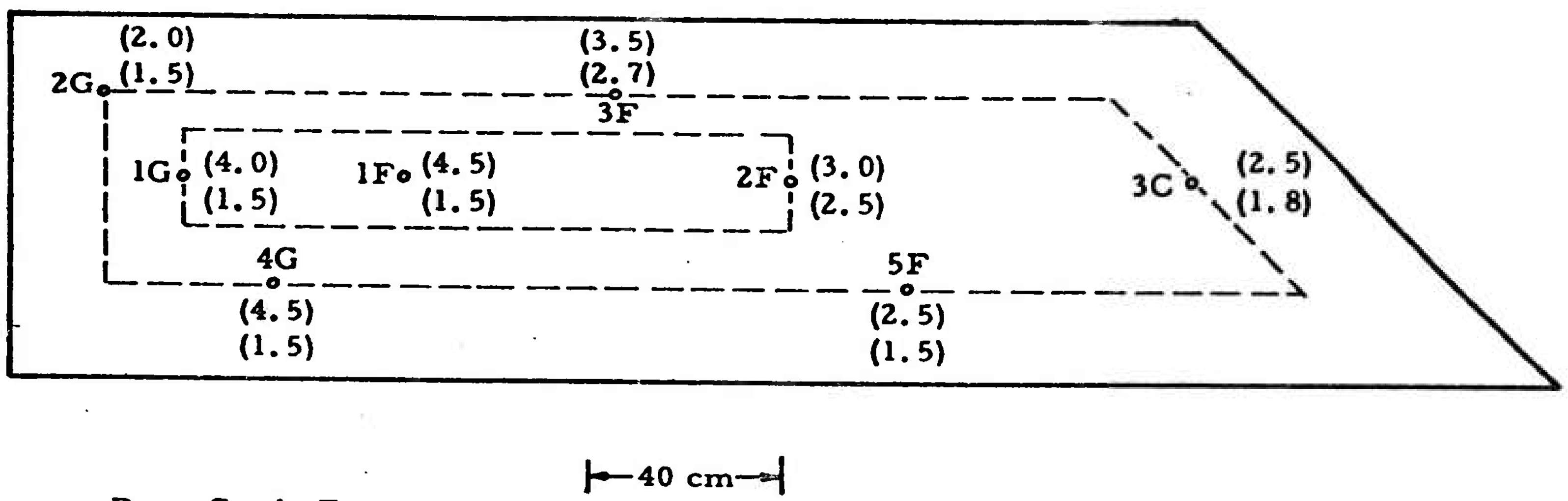
Down Grade Fast

Ring	A	u	$P=u^2A$
1	364	11.0	44,100
2	2374	14.0	468,000
3	3892	7.0	190,500
Total			702,600

In Water 0-15 mph

Ring	A	u	$P=u^2A$
1	364	7.2	18,880
2	2374	11.1	292,000
3	3892	5.6	121,800
Total			432,680

FIGURE 33. ARRANGEMENT OF SAMPLING ON DRIVER'S ENGINE ACCESS PANEL



Down Grade Fast

Ring	A	u	P=u²A
1	2,520	4.00	40,200
2	6,440	3.25	67,600
3	12,624	1.50	28,400
Total			136,200

In Water Accel, 0-15 mph

Ring	A	u	P=u²A
1	2,520	1.75	7,750
2	6,440	1.90	23,100
3	12,624	0.90	10,230
Total			41,080

FIGURE 34. ARRANGEMENT OF SAMPLING FOR LEFT ARMOR WALL.

The "sound radiating power" u^2A was also computed[7], and the "total sound radiating power" in cm^4/sec^2 was determined by adding the figures for all rings.

Table VIII gives the condensed results of the vibration survey, from this the relative sound radiating power of the vehicle panels can be compared. In this table, the most important sound-radiating surfaces, as shown by their measured "total sound-radiating power" u^2A, are listed near the top. These turn out to be mostly thin panels of low stiffness, large area, or both. It should be noted that the list does not include all of the panels in the vehicle; however, all types are included. If two panels were similar, measurements were sometimes made on only one.

5 Impulse Testing - Panel Fundamental Frequency Studies

During the vibration surveys it became clear that many of the vibrating panels had distinct resonances. Thus, impulse testing was conducted on the more important sound-radiating panels to explore their behavior more fully. The technique used was to record the vibratory response to light blows on the panel with a 2-lb rubber mallet. The recordings were made with the CEC oscillograph by the same system as used for the vibration survey. The fundamental frequencies of the panels were readily measured by placing a transducer near their center. It was possible to record the second and higher order harmonics by placing transducers at positions other than the center and striking the panels at different places. However, the higher order harmonics were damped out rapidly. Figure 35 illustrates a sample impulse test showing the fundamental frequency of the panel under test to be 39 cps. Table IX lists the natural frequencies of the important sound-radiating panels in the vehicle. The measured fundamental frequencies agree with the resonant frequencies observed during the vibration survey.

The natural damping rate of each panel was also measured and is included in Table IX in terms of its solid damping factor[8]. This is given by:

$$\gamma = \frac{1}{\pi} \ln \frac{X_n}{X_{n+1}}$$

where $\frac{X_n}{X_{n+1}}$ represents the ratio of succeeding amplitudes. All the values

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7 See Section III-C-2. Actually, the u^2A should be multiplied by a constant factor to be sound power, but for our comparative intents, u^2A was sufficient.

8 Thompson, W. R., Mechanical Vibrations, Chapter 3, Prentice-Hall, Inc. New York, New York, 1948.

TABLE VIII. SUMMARY OF VEHICLE INTERIOR VIBRATION SURVEY

Panel Description	Area, cm^2	Total Radiating Power u^2A, cm^4/sec^2 On land 25-35 mph	In water 15 mph
Side of Battery Box	1,536	6,140,000	98,300
Right Bench Back Rest	3,008	3,490,000	3,630,000
Right Front Floor Plate	12,540	2,180,000	1,060,000
Fuel Tank Wall (Tank 1/3 Full)	7,676	2,020,000	335,000
Left Front Flooring Plate	12,540	1,277,000	907,500
Engine Access Panel, Driver's	6,630	702,000	433,000
Rear Flooring Plate	8,448	581,000	101,000
Engine Access Panel, Passenger's	9,880	399,700	233,000
Total for Sub-floor	35,600	301,000	93,000
Left Armor Wall	21,584	136,200	41,000
Right Armor Wall	22,550	99,700	50,000
Engine Wall Frame, Passenger's	4,880	88,200	48,400
Outer Plate, Rear Door	7,820	74,300 (?)	86,000 (?)
Right Bench Seats	7,471	75,800	18,370
Top Armor of Left Fender	11,520	64,000	30,500
Engine Wall Frame, Driver's	5,480	57,100	28,000
Top Armor of Right Fender	7,144	56,200	25,100
Rear Top Hatch	8,208	51,200 (?)	8,200
Side Armor of Right Fender	7,336	27,300	36,530
Driver's Front Armor Wall	7,166	36,000 (?)	14,400
Inner Plate, Rear Door	7,820	17,600	31,400
Floor of Driver's Area	4,104	19,000	7,150
Rear (ramp) Wall	8,700	14,800	12,200
Engine Wall, Passenger's	2,280	13,125	4,185
Rear Wall (beside ramp)	4,452	10,000	8,420
Gas Pedal Floor Armor	2,050	4,610	1,153
Engine Wall, Driver's	1,805	1,800	1,500

NOTE: (?) means too few or badly distributed sampling points.

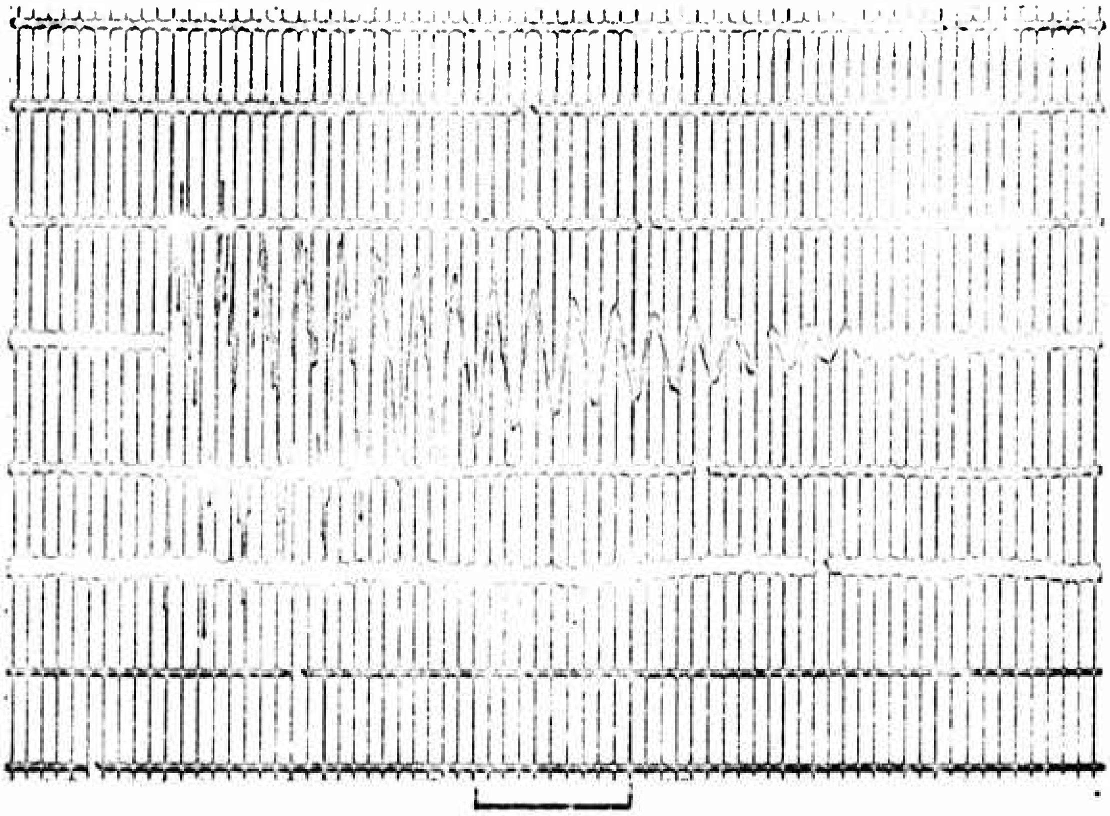


FIGURE 35. IMPULSE TEST OF RIGHT BENCH BACK-REST.

TABLE IXa. PANEL FUNDAMENTAL FREQUENCIES AND DAMPING RATES

Panel	Fundamental frequency cps	Solid damping factor γ
Engine access panel, driver's	39	.017
Engine access panel, passenger's	52	.029
Right front flooring panel (center)	47	.030
Same (front and rear)	36	
Top of battery box	66	.018
Side of battery box	100	.013
Left bench back rest	38	.005
Left bench back rest*	42	.025
Right bench back rest	39	.039
Right bench back rest*	45	.055
Fuel tank wall (1/5 full)	56	not measured
Fuel tank wall (full)	16	not measured
Left armor wall	~225	not measured
Sub-floor	~265	not measured
3rd torsion bar from rear	63	not measured

* Measurements made with channels clamped to back rests.

TABLE IXb. PRINCIPAL DRIVING FORCE FREQUENCIES AT 10 AND 20 MPH

Gear	Speed (MPH)	Track-Sprocket Engagement Frequency (CPS)	Engine Speed (RPM)	Firing Frequency (CPS)
1st	10	29.3	3660	244
2nd	10	29.3	2630	176
3rd	10	29.3	1840	123
3rd	20	58.6	3680	246
4th	10	29.3	1340	89.2
4th	20	58.6	2680	179
5th	10	29.3	960	64
5th	20	58.6	1920	128
6th	10	29.3	692	46
6th	20	58.6	1380	92

given are considered low in mechanical systems; thus, in part accounting for the high level of noise radiation of these panels at their resonant frequencies.

This results because under resonance conditions the vibration response of a panel is limited only by its damping as the stiffness and mass reactive vibratory impedances cancel. Thus, if the damping is small, the panel vibrations can become quite severe at resonance which, in turn, will radiate intense noise levels.

From observation of the range of the measured panel fundamental frequencies together with the major vehicle driving frequencies, it is evident why a noise problem exists in the vehicle. For example, one of the noted predominant driving frequencies was the track-sprocket engagement frequency which is approximately equal in cps to three times the speed of the vehicle in mph (ie, 45 cps at 15 mph). Thus, this driving frequency will cause at least one panel to be in resonance practically throughout the operating range of the vehicle.

D. Vibration Source and Transmission Path Evaluation

One of the most informative techniques that was used for identifying and evaluating vibration and sound sources was to conduct tests while selectively removing various possible sources. Thus, by comparison of the results obtained, it was possible to evaluate the importance of each source. To study the relative source strengths, measurements were made of the sound at the standard positions while operating at selected vehicle speeds. It was felt that the measurement of the end product of the vehicle panel vibrations (ie., airborne sound) would give the best qualitative estimate of relative importance of the various sources. Measurements of sound level near the center of the compartment will afford a more correct weighting of relative source strength than the other possibilities available, because the location nearly equidistant from the radiating panels will minimize localized panel effects as mentioned previously. It would be practically impossible to measure the vibrations of each element in the vehicle for each source alteration test, and the selection of a few positions could quite possibly lead to erroneous results.

However, vibration measurements were recorded on a few of the stiffer panels during much of the testing to supplement the noise data and for transmission path studies. The stiffer elements were chosen so that slight differences in speed from test to test would not result in large differences in measured vibration amplitudes, as would be the case for less stiff panels more susceptible to strong resonances.

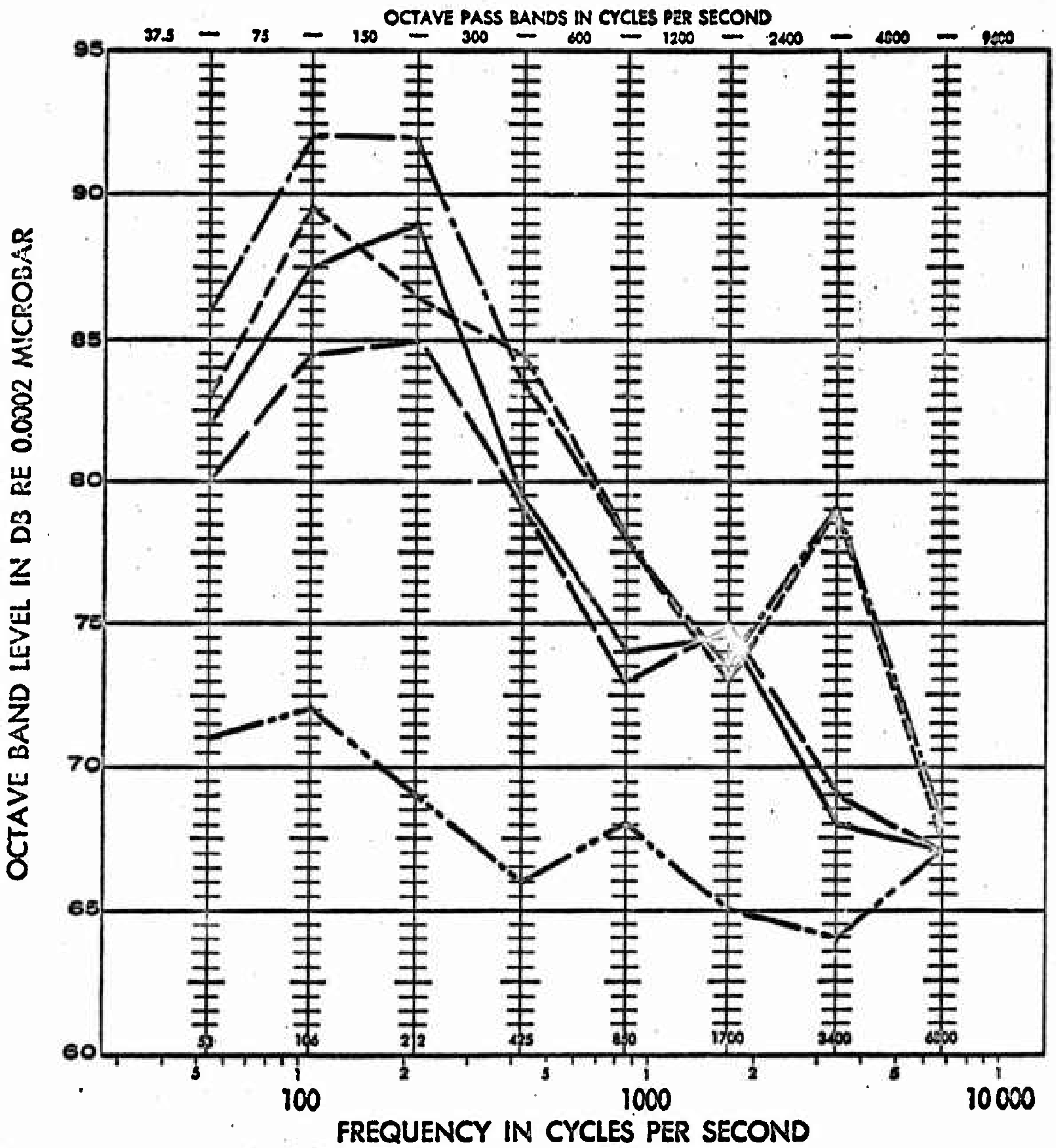
As for previous testing, the test course was the airport runway described earlier, except when it was desired to compare the effect of terrain differences. Both sound and vibration were recorded as described in Section II-B. In addition, some vibration measurements were recorded with the light beam oscillograph. The results were analyzed, both by comparing overall amplitudes and by studying the frequency spectra. Octave band analyses of the taped data were made for studying the general nature of the spectra. For detailed analyses of the spectra to correlate frequency components with vibration sources, a Panoramic Radio Products Co. Type LF-2 subsonic analyzer and a Panoramic Radio Products Co. Type LP-1a sound analyzer were used.

Most of the testing for vibration source evaluation was done while operating with the driver's hatch opened and all other hatches closed. The driver's hatch was left open for convenience and for signaling purposes during the towing tests. It was decided to use one low vehicle speed and one high speed for testing. Initially, speeds of 10 mph and 25 mph were chosen; however, difficulty was encountered in towing the vehicle at a speed of 25 mph, and hence for these tests a speed of 20 mph was used.

During this evaluation all tests on an individual source were conducted on the same day or within as short a span of time as possible. This minimized errors arising from changes in operating conditions. The testing procedure for source identification was initiated by studying the effects produced by simple modifications such as adding an external muffler, using rubber idler wheels, and varying the track tension. The next step was to conduct tests with various sources removed such as towing tests, vehicle on blocks tests, and tests with the tracks removed. Then finally specific testing was conducted to study the track reactions with the drive sprockets, idler wheels, and road wheels. These tests and their results are discussed in detail in the following sections.

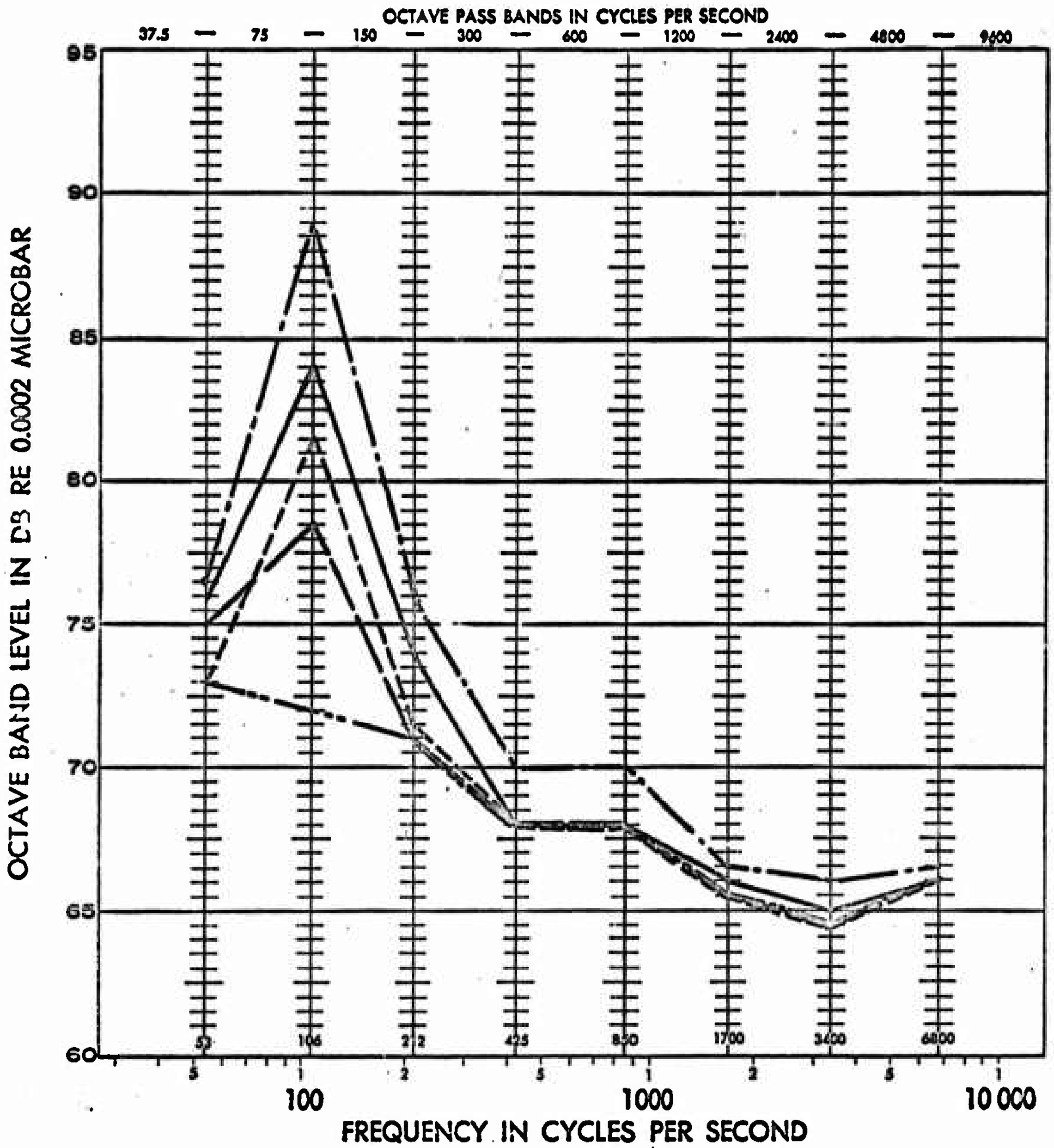
1. External Muffler Effects

Early in the testing period, two external mufflers were obtained and tested on the vehicle for noise reduction. The first muffler (No. A-1) was used only temporarily until the better muffler (No. A-2) was obtained. The A-2 muffler was then used throughout most of the testing period. The attenuation afforded by this muffler under engine idling conditions is given in Figure 36 for internal noise and Figure 37 for external noise. There was an attenuation of 3 db in the overall internal noise level at all speeds of measurement. These reductions were caused by important suppression in the low frequencies as evident in Figure 36. Reductions in external engine noise under idle conditions measured at a distance of 50 feet from the vehicle



1500 RPM, WITHOUT EXTERNAL MUFFLER
1500 RPM, WITH EXTERNAL MUFFLER
2500 RPM, WITHOUT EXTERNAL MUFFLER
2500 RPM, WITH EXTERNAL MUFFLER
AMBIENT NOISE

FIGURE 36. INTERNAL SPECTRA WITH ENGINE IDLING SHOWING MUFFLER EFFECT, ALL HATCHES CLOSED.



——— 1500 RPM, WITHOUT EXTERNAL MUFFLER
— — 1500 RPM, WITH EXTERNAL MUFFLER
—·— 2500 RPM, WITHOUT EXTERNAL MUFFLER
- - - - 2500 RPM, WITH EXTERNAL MUFFLER
—··— AMBIENT NOISE

FIGURE 37. EXTERNAL SOUND SPECTRA WITH ENGINE IDLING SHOWING MUFFLER EFFECT, 50 FT FROM VEHICLE.

ranged from an almost negligible amount at low speed to a quite substantial reduction of 7 db at 2500 rpm. Again, as evident in Figure 37, the major attenuation observed was the low frequencies; however, there was a noticeable effect at high frequencies at higher engine speeds.

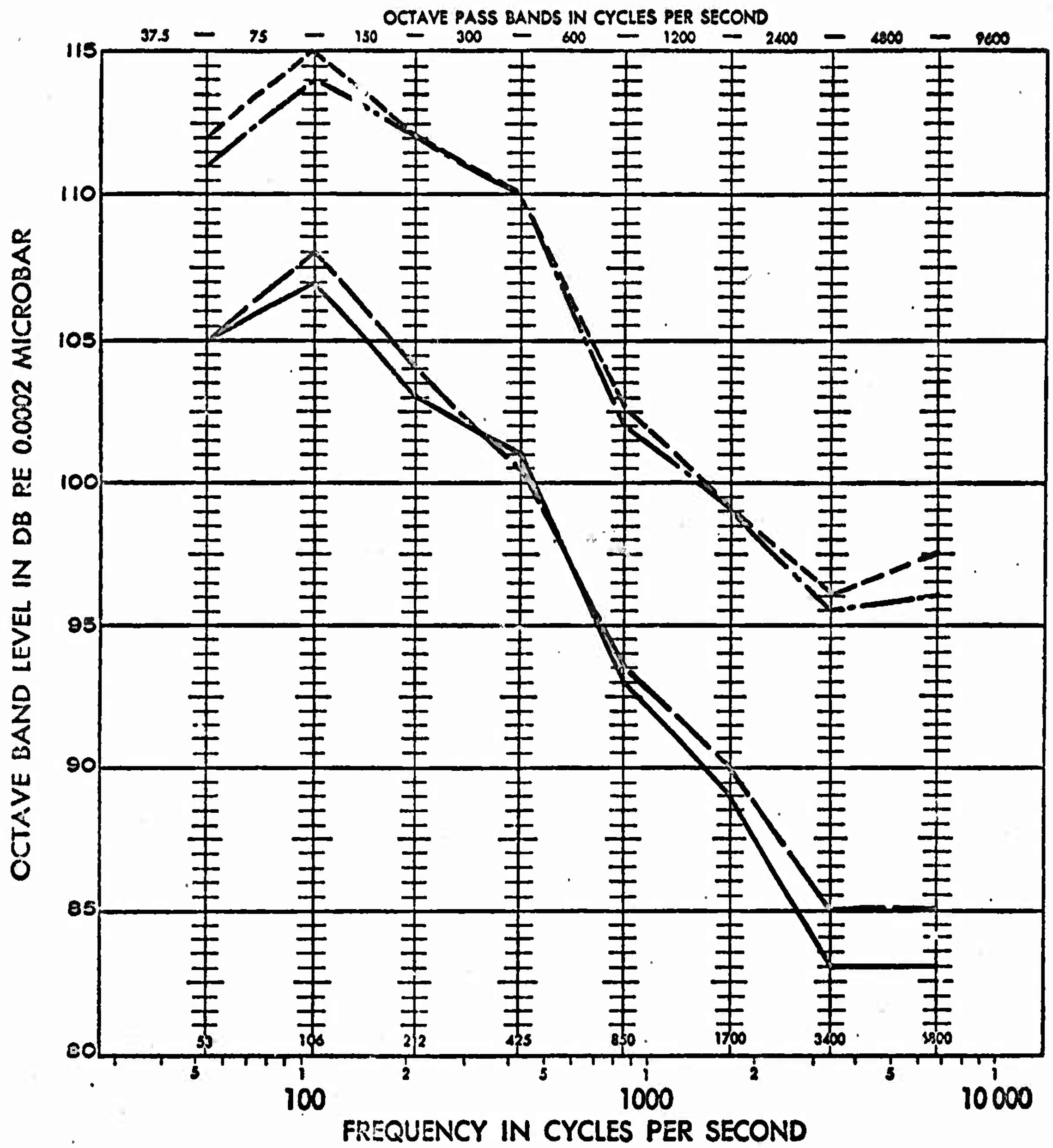
The effects produced by the A-2 muffler with the vehicle in motion were almost negligible. Inside the vehicle there was almost no difference between operating with the muffler in place or without it as shown in Figure 38. A slightly higher inside noise is indicated with the muffler in place, presumably because the exhaust was then directed over top the vehicle. However, there was a slight reduction in external noise at low speed with the muffler on the vehicle as illustrated in Figure 39. It is also indicated that there was a slight suppression of high frequencies at both vehicle speeds.

2. Rubber Idler Wheel Effects

A pair of rubber idler wheels were obtained, installed, and tested for their vibration damping effectiveness. From Figure 40 it is evident that the rubber idlers produced a substantial reduction in the internal noise level, especially at low speeds. The decrease in overall level tapered from 5 db at 10 mph to 1 or 2 db at 25 mph. Vibration data was also taken on various panels comparing the rubber idlers to the steel idlers during the coasting test procedure. Results tabulated in Table VII indicate a slight reduction in vibrational velocity of the panels with the rubber idlers on the vehicle. Additional comparisons were made with the vehicle blocked up. These will be discussed in a later section. All the tests indicate that the rubber idlers do produce a reduction in vibration and, therefore, in radiated noise levels.

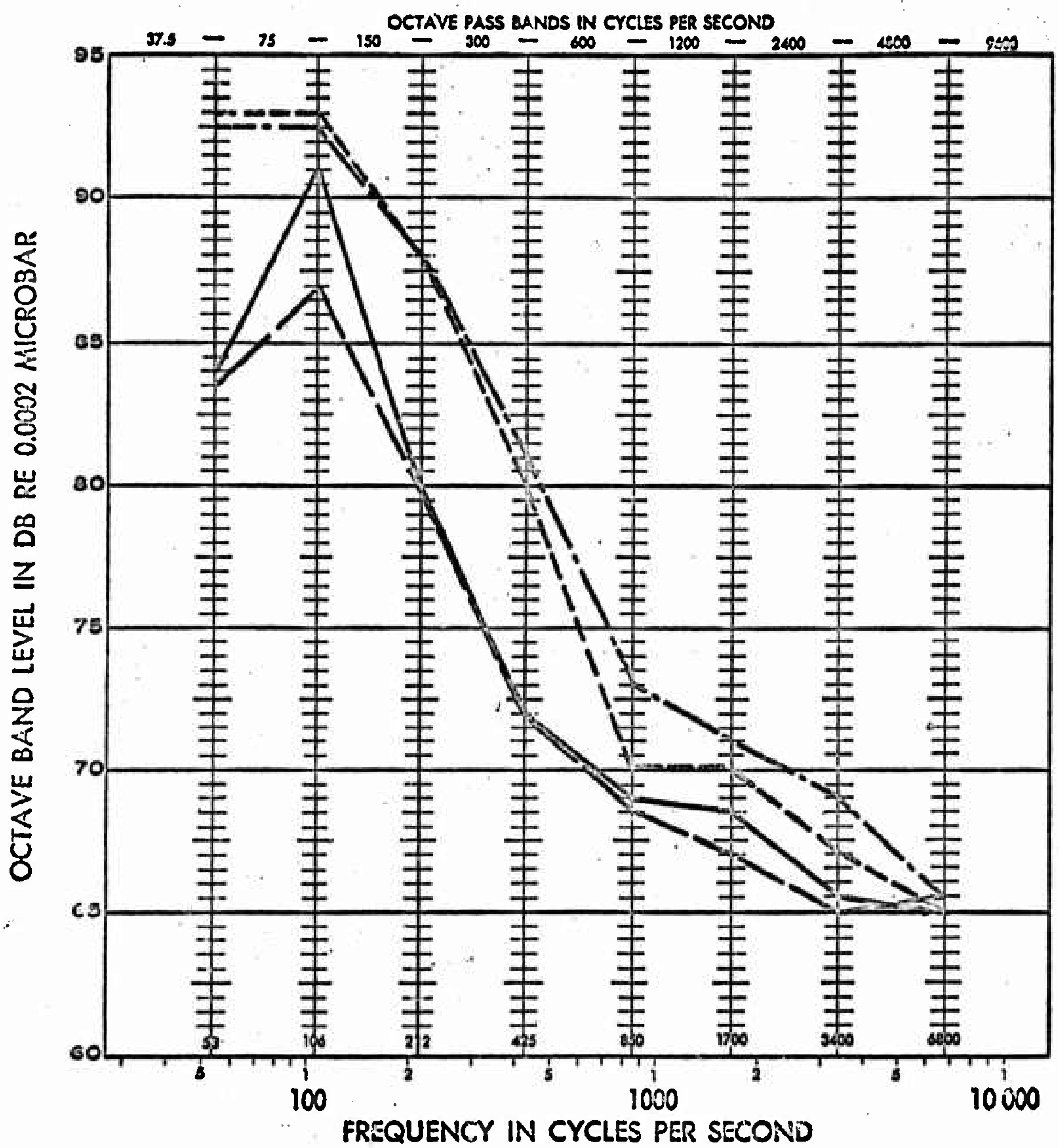
3. Track Tension Effects

Testing was conducted to evaluate the variation of vibration and noise levels as a function of track tension. The internal noise was recorded at vehicle speeds of 10, 20, and 25 mph for three different track tensions. These track tensions used were (1) recommended tension; (2) approximately 1/2 the recommended value (ie, removing 1/2 of the grease from the tension cylinders), and (3) with all of the grease removed. Octave bands of the internal noise for various track tension are given in Figures 41 and 42 for 10 and 25 mph, respectively. There was an overall reduction of 4 db noted at all speeds by reducing the tension to its minimum value. Tests varying the tension with the vehicle blocked up gave similar results. Also, vibration data with the vehicle blocked up showed reduced levels



10 MPH, WITHOUT EXTERNAL MUFFLER
10 MPH, WITH EXTERNAL MUFFLER
25 MPH, WITHOUT EXTERNAL MUFFLER
25 MPH, WITH EXTERNAL MUFFLER

FIGURE 38. INTERNAL SOUND SPECTRA SHOWING MUFFLER EFFECT, ALL HATCHES EXCEPT DRIVER'S CLOSED.



10 MPH, WITHOUT EXTERNAL MUFFLER
10 MPH, WITH EXTERNAL MUFFLER
20 MPH, WITHOUT EXTERNAL MUFFLER
20 MPH, WITH EXTERNAL MUFFLER

FIGURE 39. EXTERNAL SOUND SPECTRA SHOWING MUFFLER EFFECT, 50 FT FROM VEHICLE.

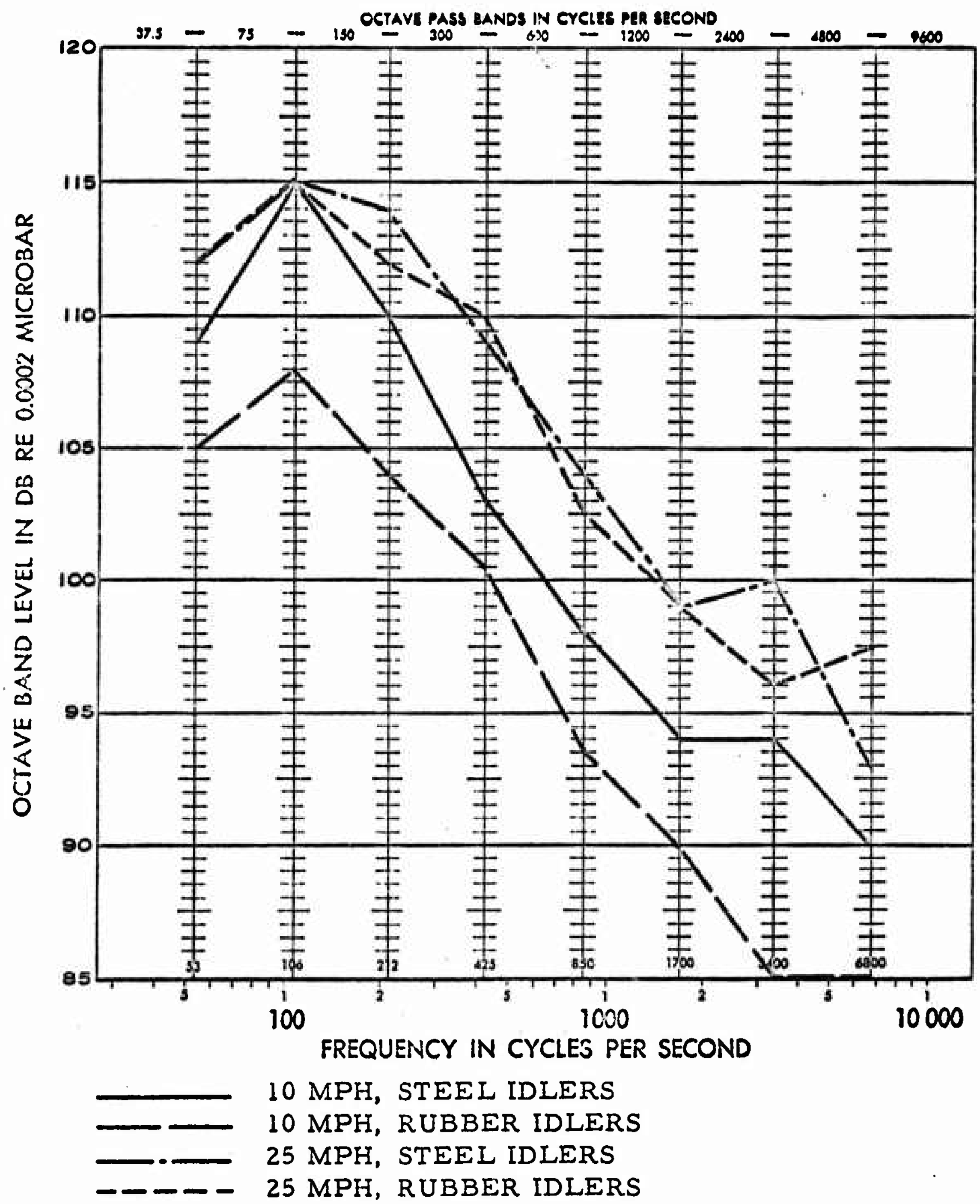
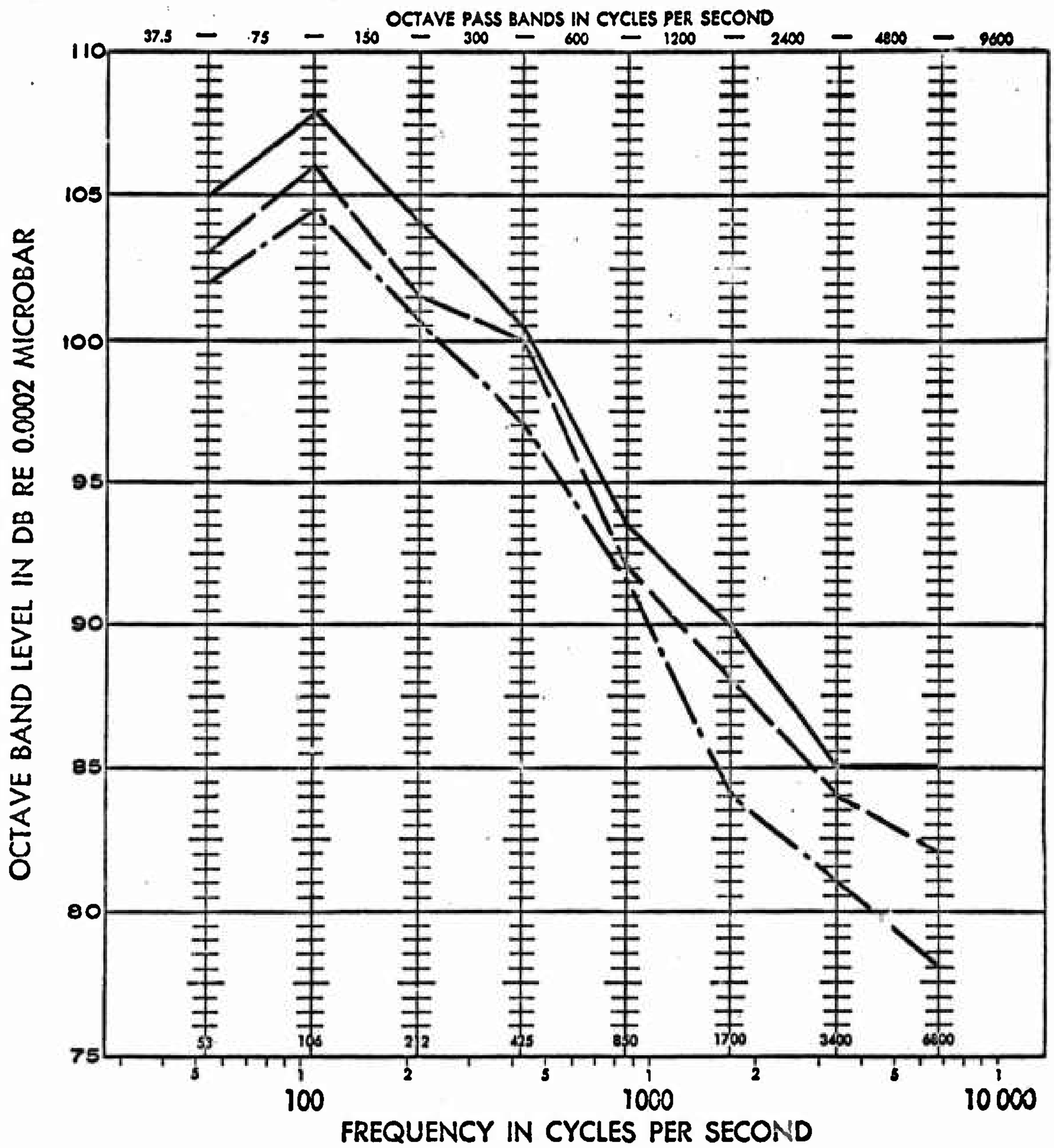


FIGURE 40. INTERNAL NOISE SPECTRA COMPARING STEEL TO RUBBER IDLERS.

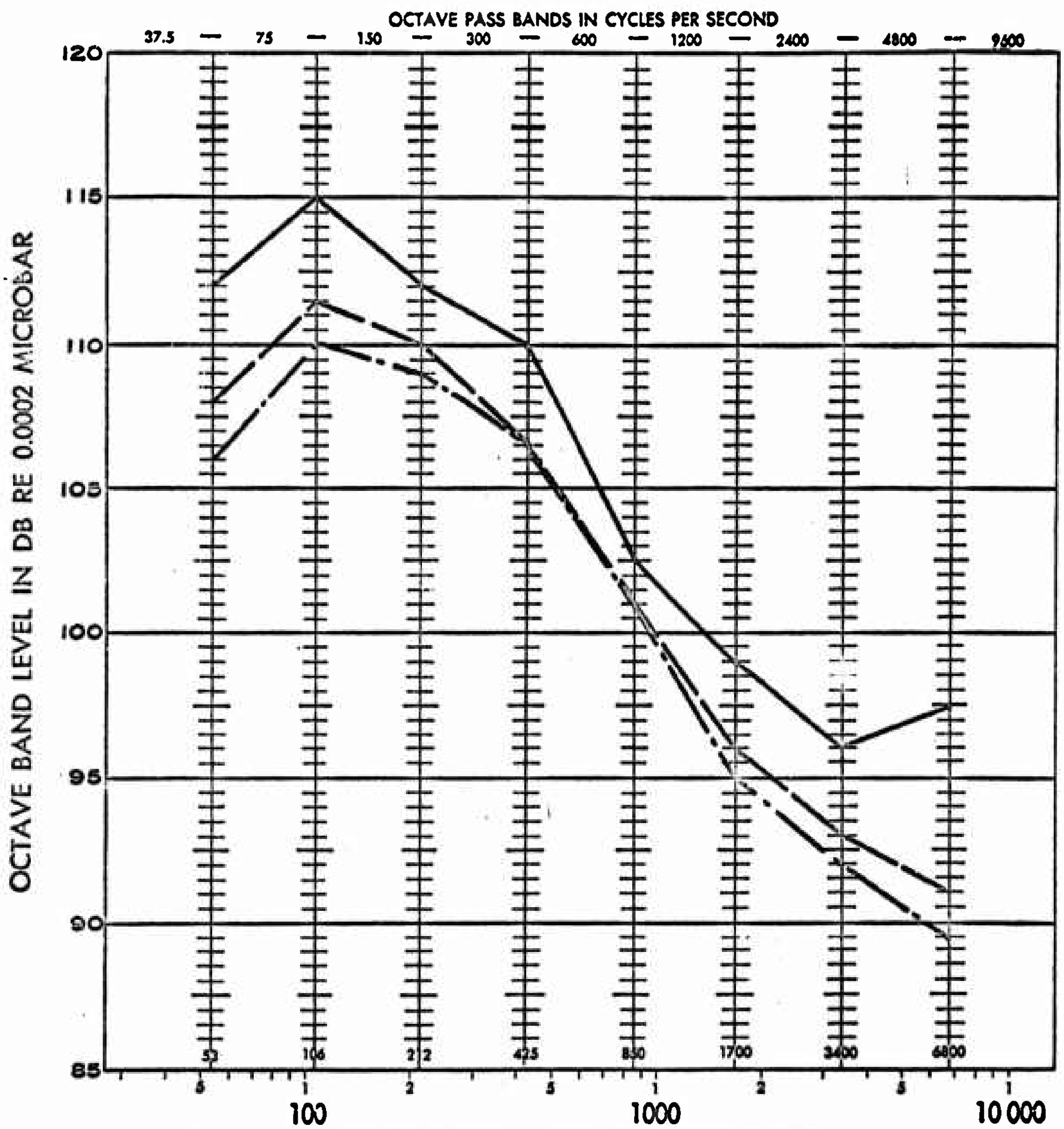


PROPER TRACK TENSION

1/2 PROPER TRACK TENSION

0 TRACK TENSION

FIGURE 41. INTERNAL NOISE SPECTRA AT 10 MPH SHOWING TRACK TENSION EFFECT.



PROPER TRACK TENSION
1/2 PROPER TRACK TENSION
0 TRACK TENSION

FIGURE 42. INTERNAL NOISE SPECTRA AT 25 MPH SHOWING TRACK TENSION EFFECT.

when tension was reduced. Thus, track tension has an important effect on the levels of noise and vibration created by the vehicle.

4. Selective Source Removal Test Procedure

The following testing procedure which in effect selectively removes different sources was evolved for evaluating the contributions of the more important vibration sources. These tests involved features such as towing the vehicle, removing the tracks, and operating blocked up off the ground. Measurements were made of the sound at the center of the passenger compartment and the vibrations of several of the stiffer panels including the armor wall, track fender top, engine wall, and hull bottom.

a. Operating the vehicle with the tracks on:

(1) Driving normally, thus determining the actual noise and vibration conditions.

(2) Towing the vehicle with the engine running at an rpm comparable to that encountered when driving at the same speed, thus eliminating the engine loading and track loading effects.

(3) Towing with the engine off, eliminating the engine noise entirely

b. Operating with the tracks removed: these tests eliminated the reactions between the tracks and the sprockets, idlers, and road wheels.

(1) Towing with the engine running and the transmission in gear operating at a speed comparable to the actual vehicle speed.

(2) Towing with the engine running and transmission in neutral, eliminating transmission noise in addition to track reactions.

(3) Towing with the engine off to also eliminate engine noise.

(4) Towing on roads of different textures for suspension system contribution studies.

(5) Operating the vehicle with the engine running and transmission in gear, eliminating road noise in addition to track reactions.

(6) Operating with the engine idling and transmission in neutral, eliminating all sources but the unloaded engine.

c. Operating with the vehicle blocked up (ie, tracks free to turn without touching the ground): These tests eliminated the track-ground reaction.

(1) Operating with both tracks turning.

(2) Operating with the right track locked, thus, attempting to reduce the contributions from the track reactions by one-half.

The condensed results, comparing overall levels, are given in Tables X and XI along with the results of other testing procedures. All of the tests in this section were conducted with the A-2 muffler and the rubber idlers on the vehicle. In addition to measuring overall levels, octave band and narrow band analyses were made. The Panoramic subsonic and sonic analyzers were used for plotting the frequency spectra. The spectra were studied in detail to identify pure tone components which might be missed in octave band studies. It was found that little need existed for analyzing above 1000 cps, since the levels of the spectra at higher frequencies were well below the low frequency amplitudes.

From Table X and Figures 43-45 relatively little difference is indicated between the driving and towing tests with the tracks on the vehicle. This signifies that engine vibration is almost negligible compared to the other vibration sources while moving. The observed spectra are characterized by the track-sprocket engagement frequency and additional components corresponding to multiples of this frequency. This frequency in cps is approximately equal to the vehicle speed in mph multiplied by 3. For example, at 20 mph the track sprocket engagement frequency would be 60 cps. The harmonic components existed because the track reactions are not pure tone. The internal sound spectra are further complicated by standing-wave components and contributions corresponding to the various panel resonant frequencies. Samples of the vibration spectra of some of the stiffer panel elements are presented in Figures 46-49, illustrating spectral distributions made of strong low frequency components corresponding to the major driving frequencies (track-sprocket engagement frequency and its lower harmonics) plus higher frequency components corresponding to the natural frequencies of the panels.

The vibration data as well as the sound data were analyzed using a logarithmic scale so that a greater amplitude range could be covered. Thus, using the vibration calibration signal of 1.56 cm/sec as a reference level, an arbitrary vibration "db" scale was set up. So that the correlation between

TABLE X. SUMMARY OF TEST VARIATIONS FOR THE EVALUATION OF GENERATION SOURCES AND COUPLING PATHS BY SELECTIVE ELIMINATION

Test Condition	Purpose or Effect Under Study	Internal Noise SPL, db 10 mph	20 mph	25 mph
Driving	set standard conditions	117	121	121
Driving, paved road	road-track reaction	116	120	121
Driving, soft dirt road	road-track reaction	116	120	120
Driving, rubber idlers	track-idler reaction	112	116.5	119
Driving,* A-2 external muffler	muffler effect	112-114	117-118	119-121
Driving, 1/2 track tension	track tension effect	110	113.5	116
Driving, 0 track tension	track tension effect	108	113	115
Driving, 3-6 gear range**	engine loading	113	118	119
Driving, 1-2 gear range	engine loading	112		
Towing, engine running	eng. & track loading	113-114	118	
Towing, eng. off	eng. noise	112	118	
Towing, tracks off, eng. run., in gear	track noise	99	106	
Towing, tracks off, eng. off	track & eng. noise	99	105	
Towing, tracks off, rough road	suspension system noise	100	108	
Towing, tracks off, smooth road	suspension system noise	96	103	
Standing still, tracks off, eng. run., in gear	eng. and transmission noise	94	96	
Standing still, eng. idling	eng. noise only	93	94	
On blocks, steel idlers	road-track reaction	118	120	122
On blocks, rubber idlers	track-idler reaction	110-113	115-116	119-122
On blocks, right track locked	vibration transmission from track	111	112	118
On blocks, rubber idlers with spacers	"rubbing" noise caused by track on inside of idler	111		120
On blocks, steel idlers, rubber cushions removed from sprockets	removing all cushioning from sprockets & idlers	117		126
On blocks, rubber idlers, pads glued to sprocket teeth	eliminate metal-to-metal contact between track & sprocket	112		122
On blocks, sprocket teeth removed	eliminate metal-to-metal contact between track & sprocket	112		121
Driving, idler wheels removed	eliminate idler-track reaction	106	113	116

* All following tests with rubber idlers and A-2 external muffler unless otherwise noted.
** All tests unless otherwise noted in 3-5 gear range.

TABLE XI. VIBRATION LEVELS FOR GENERATION SOURCE AND COUPLING PATH EVALUATION STUDIES

	Total RMS Vibration Velocity, db*					
	Left wall		R. fender top		Engine wall (pass. comp.)	
	10 mph	20 mph	10 mph	20 mph	10 mph	20 mph
Driving	110	113	114	121	112	120
On blocks**	112	117	112	122	115	124
On blocks, R. track locked**	107	112	109	113	112	117
Towing, eng. running	111	114	112	122	113	121
Towing, eng. off	111	114	112	116	113	118
Towing, tracks off, rough road, eng. run.	101	108	103	111	102	109
Towing, tracks off, rough road, eng. off	99	108	103	109	101	108
Towing, tracks off, smooth road, eng. run.	94	102	98	107	101	105
Towing, tracks off, smooth road, eng. off	93	101	97	104	96	102
Standing still, tracks off, eng. run., in gear	93	92	90	93	98	102
Standing still, tracks off, eng. idling	93	92	90	91	97	101

* 1.56 cm/sec set arbitrarily to 120 db
** Tests at 10 and 25 mph

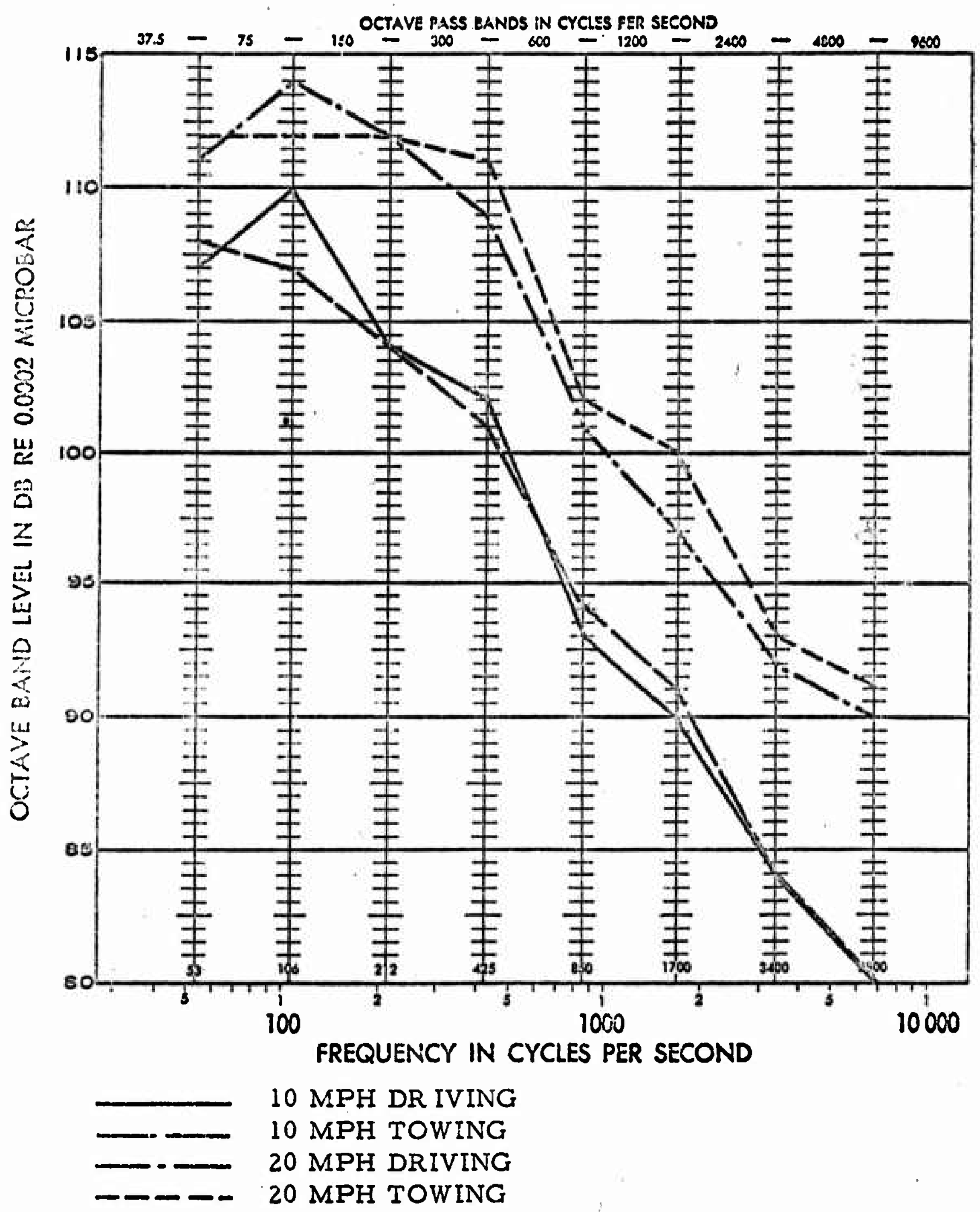


FIGURE 43. INTERNAL SOUND SPECTRA, COMPARING DRIVING TO TOWING.

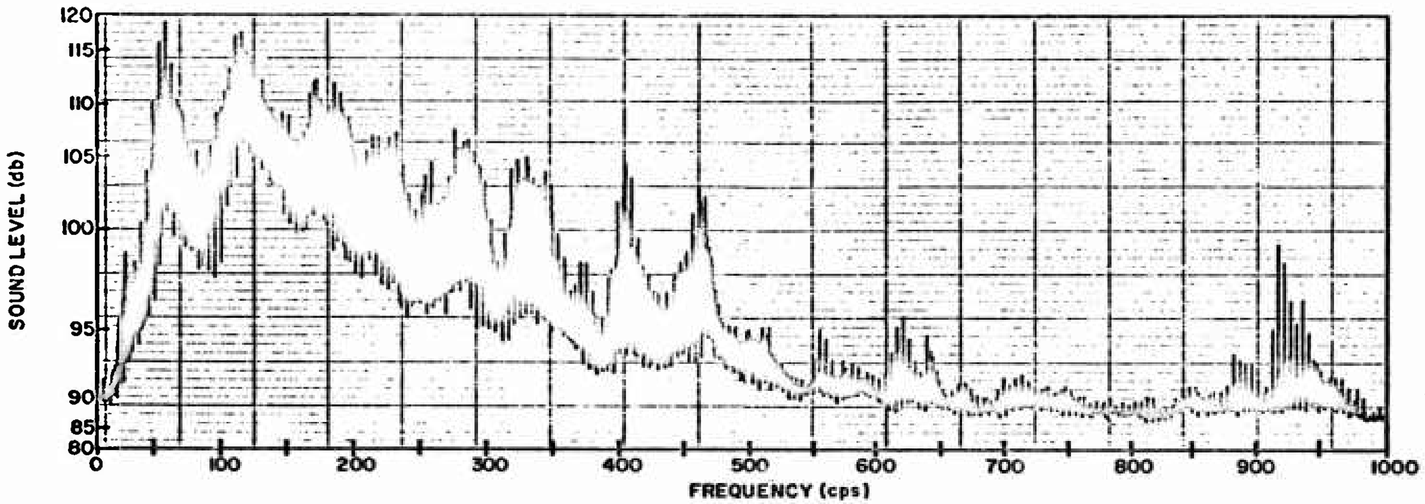


FIGURE 44
INTERNAL SOUND SPECTRUM DRIVING AT 20MPH.

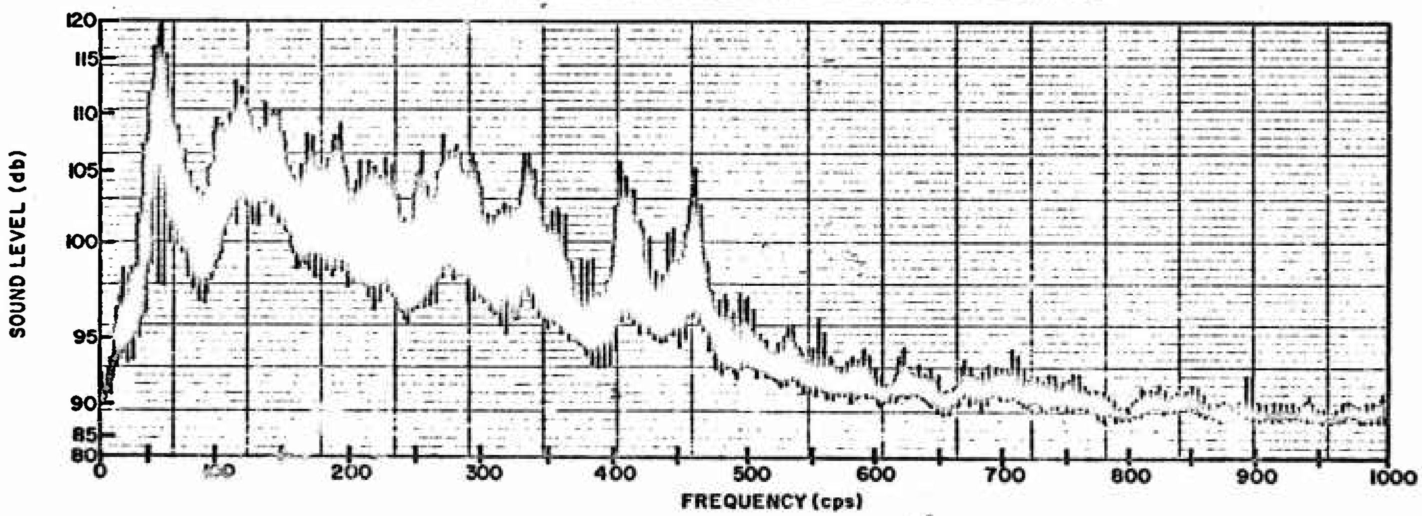


FIGURE 45
INTERNAL SOUND SPECTRUM TOWING VEHICLE WITH
WITH THE ENGINE OFF AT 20 MPH.

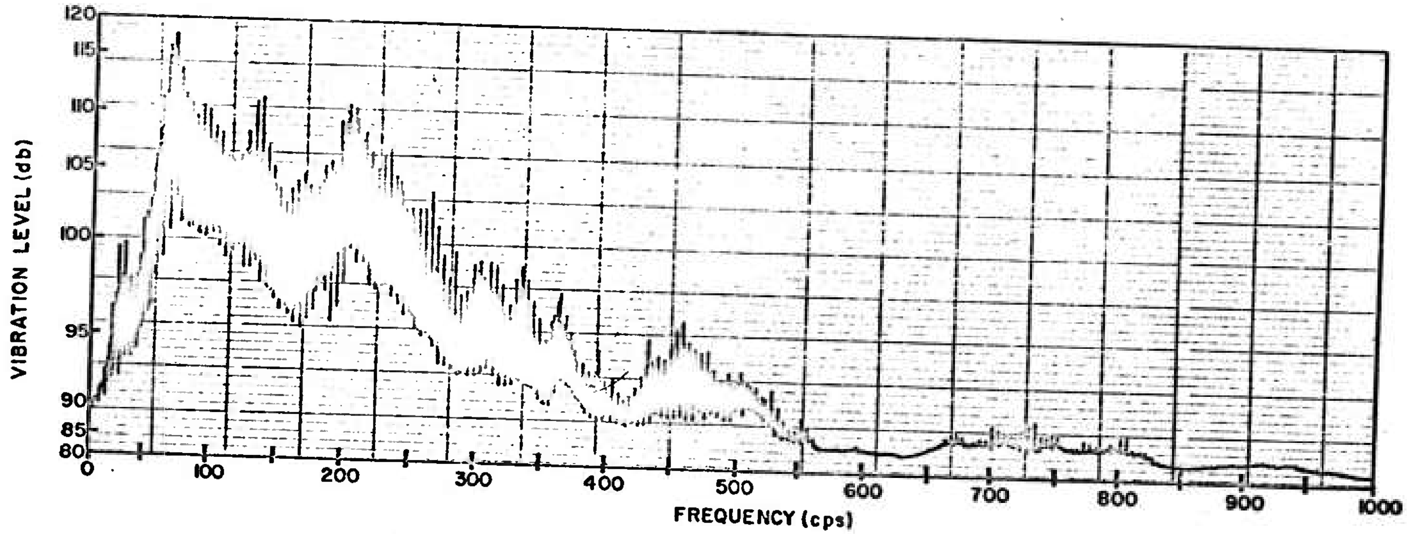


FIGURE 46
VIBRATION SPECTRUM OF LEFT ARMOR WALL, DRIVING AT 25 MPH.
AMPLITUDE REFERENCE LEVEL: 1.56 cm/sec = 120 db.

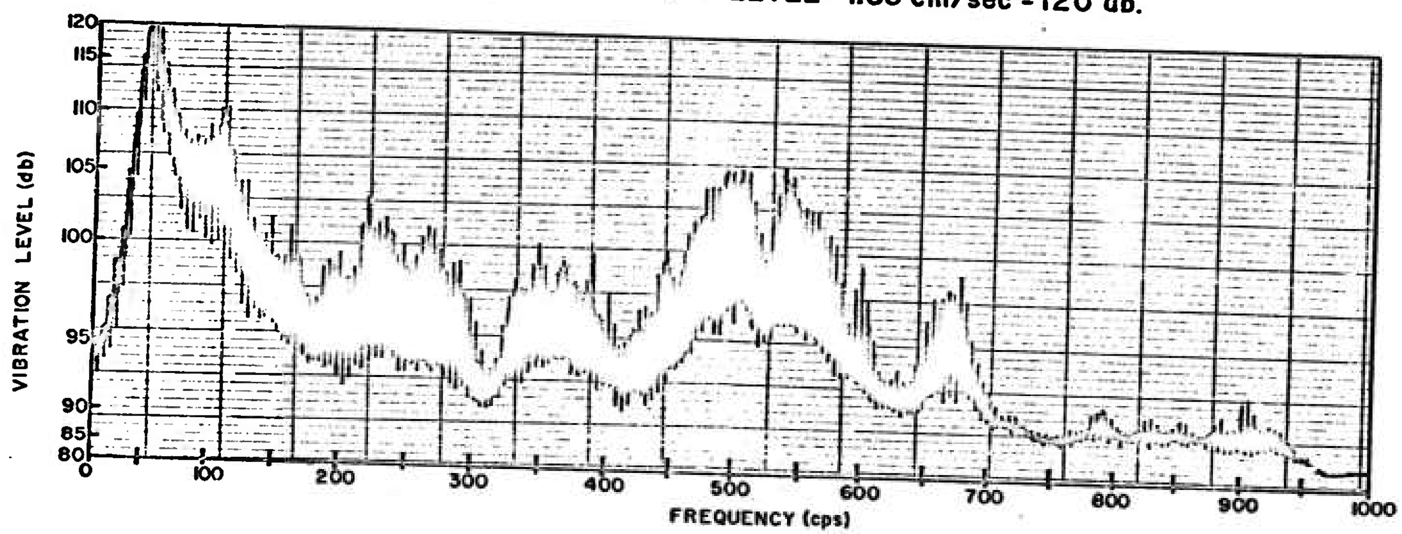


FIGURE 47
VIBRATION SPECTRUM OF TOP ARMOR OF RIGHT FENDER, DRIVING AT 20 MPH.
AMPLITUDE REFERENCE LEVEL: 1.56 cm/sec = 120 db.

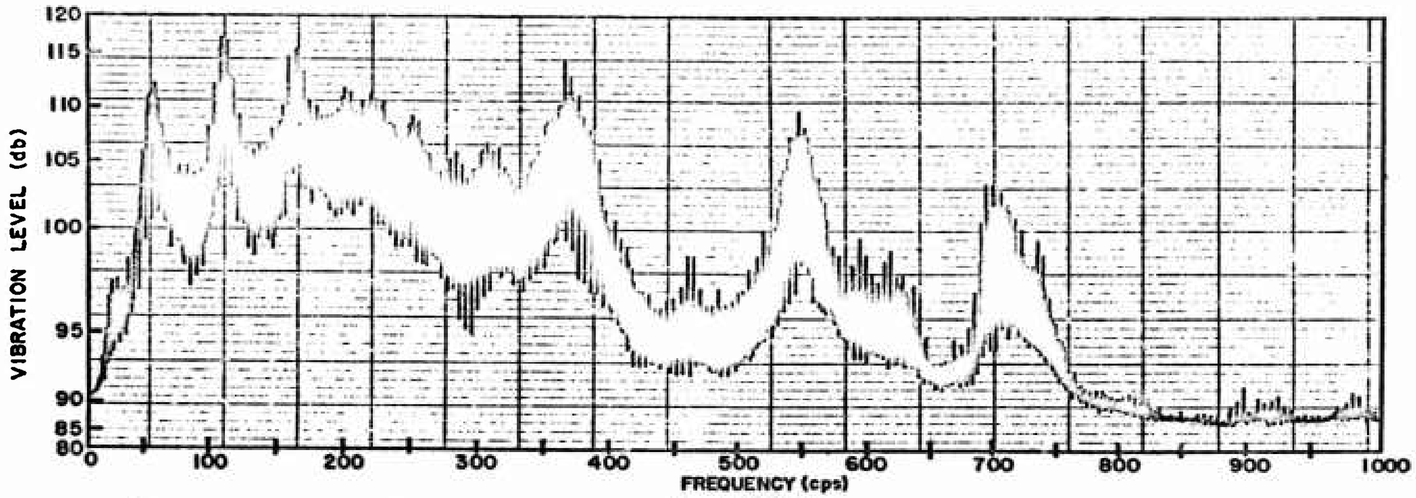


FIGURE 48

VIBRATION SPECTRUM OF ENGINE WALL (PASSENGER COMPARTMENT), DRIVING AT 20 MPH. AMPLITUDE REFERENCE LEVEL: 1.56 cm/sec = 120 db

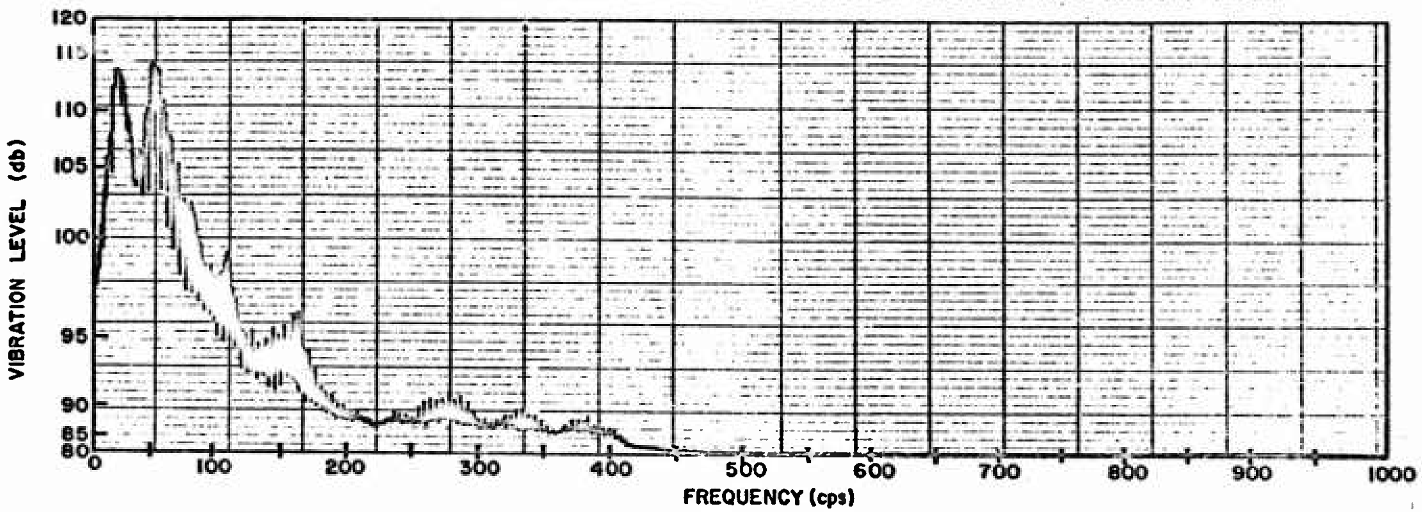


FIGURE 49

VIBRATION SPECTRUM OF BOTTOM HULL OF VEHICLE NEAR THE RIGHT REAR ROAD WHEEL SUPPORT ARM, DRIVING AT 10 MPH. AMPLITUDE REFERENCE LEVEL: 1.56 cm/sec = 120 db

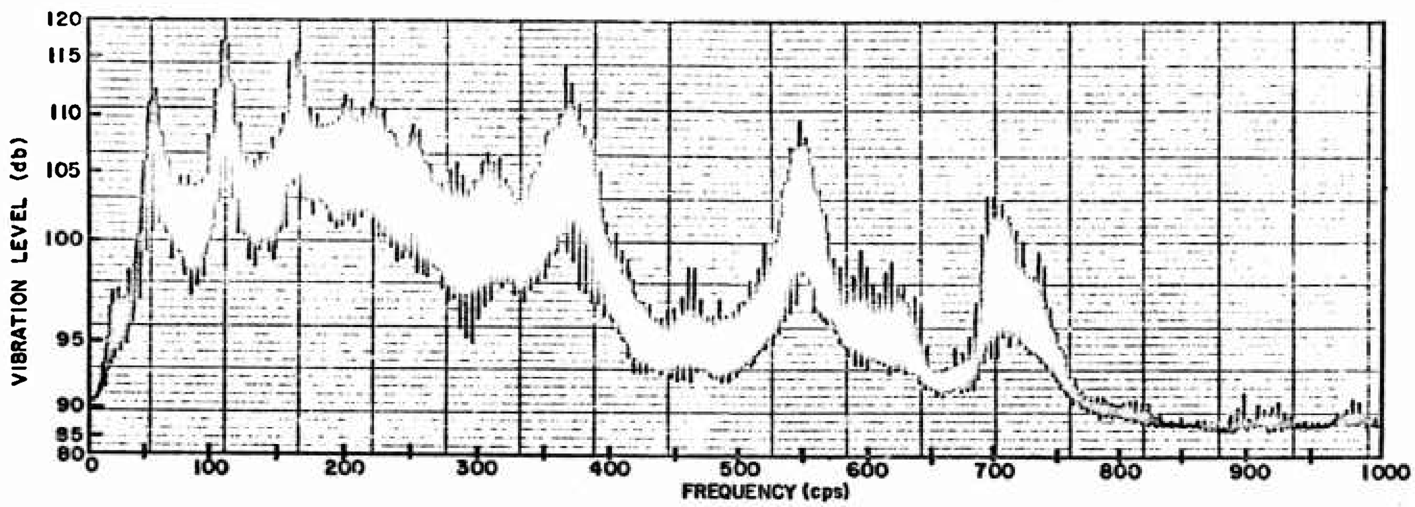


FIGURE 48

VIBRATION SPECTRUM OF ENGINE WALL (PASSENGER COMPARTMENT), DRIVING AT 20 MPH. AMPLITUDE REFERENCE LEVEL: 1.56 cm/sec = 120db

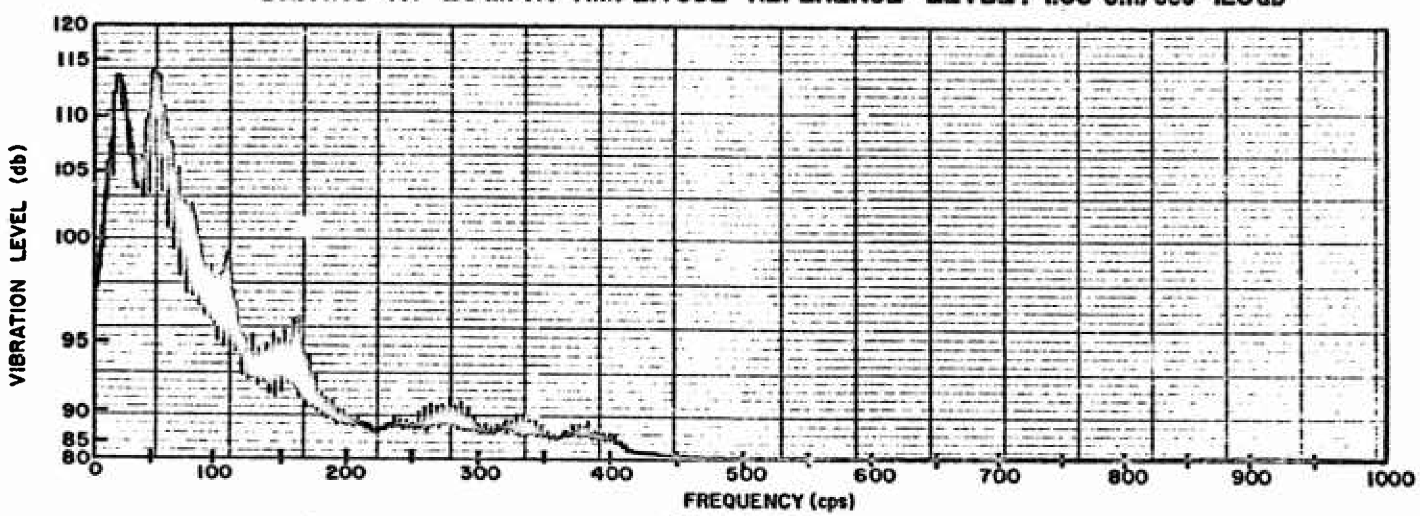


FIGURE 49

VIBRATION SPECTRUM OF BOTTOM HULL OF VEHICLE NEAR THE RIGHT REAR ROAD WHEEL SUPPORT ARM, DRIVING AT 10 MPH. AMPLITUDE REFERENCE LEVEL: 1.56 cm/sec = 120 db

sound and vibration could be noted as operating conditions were varied, the calibration level which was representative of the actual vibration measurements was set equal to 120 "db" on the logarithmic vibration scale. Since the sound pressure calibration level was 120 db and the measured test noise levels were near this value, it was possible to effectively compare noise and vibration changes brought about by altering the operating conditions.

It is indicated from Table X that there is little difference in observed noise levels between operating on blocks and driving on the ground. The frequency spectra shown in Figures 50 and 51 indicate that the same components are present; however, the higher harmonics of the track-sprocket engagement frequency appear stronger relative to the fundamental frequency for the vehicle blocked up test in comparison to the driving test.

Thus, it is indicated from these tests that the road-track produces only a minor effect on the internal noise level. This is also substantiated by the data in a previous section which illustrated the relatively small effect of driving on different road textures.

Removing the tracks from the vehicle resulted in a decrease of 12 to 15 db in the internal noise level at all test speeds. Thus, it is indicated that by far the major vibration and noise generating sources are the tracks and their reactions. With the tracks removed, the suspension system (ie, road noise) is the predominant source of vibration and radiated noise. This is evident from Table X comparing results of the towing tests without the tracks over different road textures with and without the engine running and the transmission engaged. There is a considerable variation in noise level produced by operating on various terrains; however, the engine and power train produced only minor differences. The spectra as illustrated in Figures 52 and 53 are predominantly low frequency road noise, but engine noise components are noticeable at 150 cps and more significantly at 360 cps.

Operating the vehicle standing still with the tracks removed eliminated all vibration sources except the unloaded engine and power transmission system. A high frequency sound resonance was observed with the engine operating from 2000 rpm to 2400 rpm. This resonance is illustrated in Figure 54 as a peak at about 3000 cps. The amplitude of the peak, however, is less than 80 db.

5. Vibration Transmission From the Tracks to the Vehicle

Since it was indicated from our selective source removal technique that the tracks and their reactions with the vehicle were the major sources of vibration and noise, it was decided that further investigation on the specific track reactions was necessary. These track reactions occur with the drive

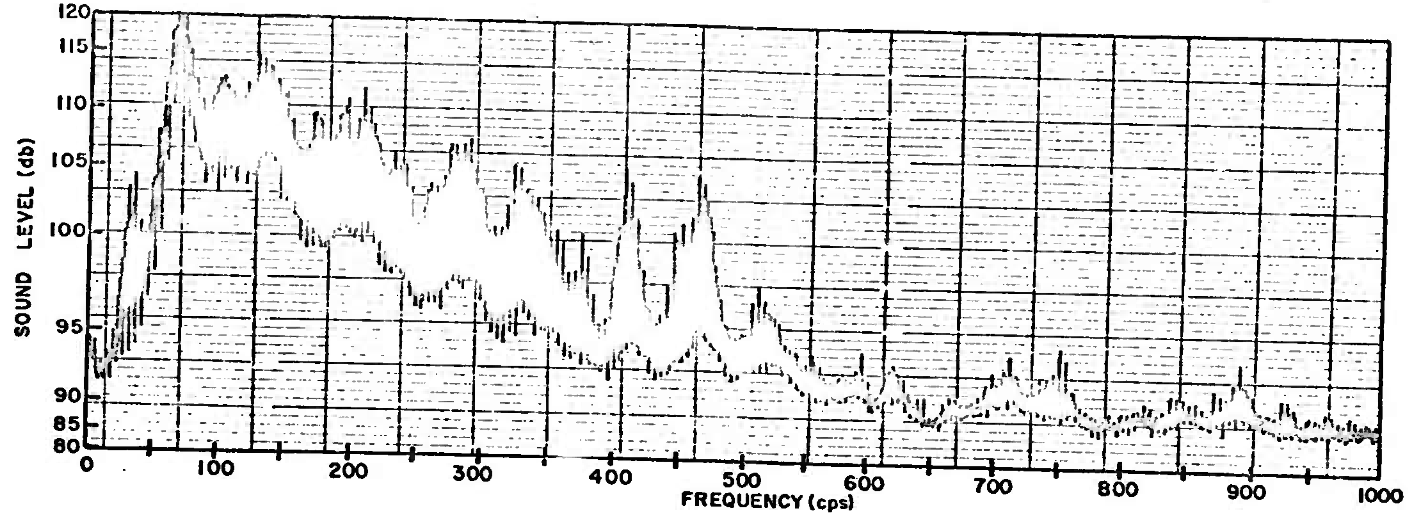


FIGURE 50
INTERNAL SOUND SPECTRUM, DRIVING AT 25 MPH.

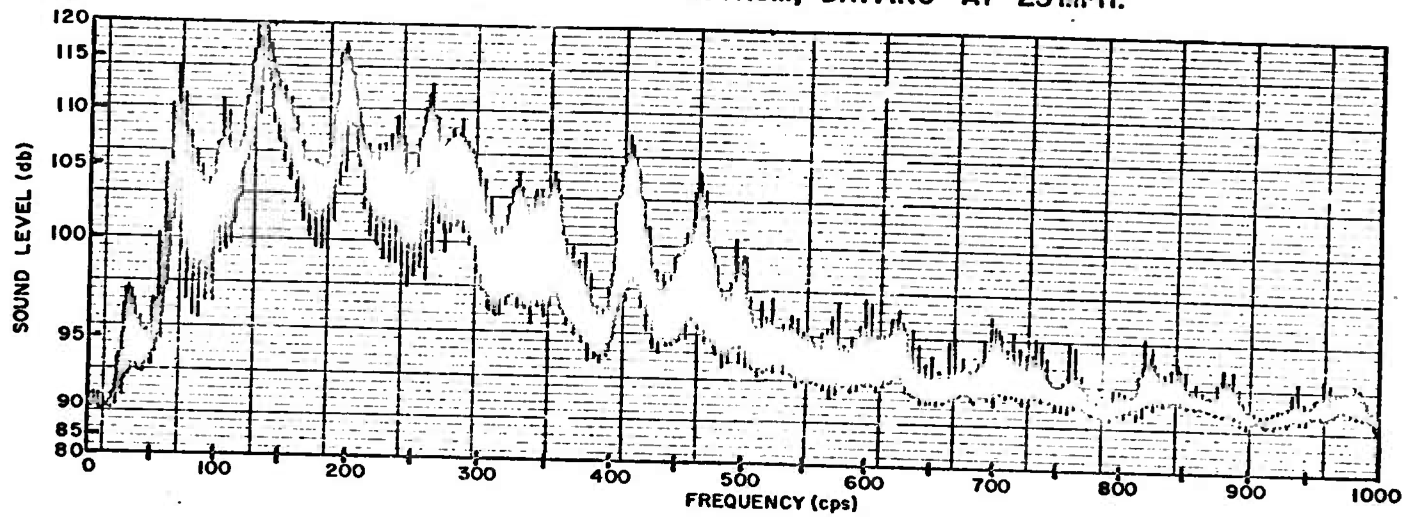


FIGURE 51
INTERNAL SOUND SPECTRUM, OPERATING WITH VEHICLE BLOCKED UP AT 25 MPH.

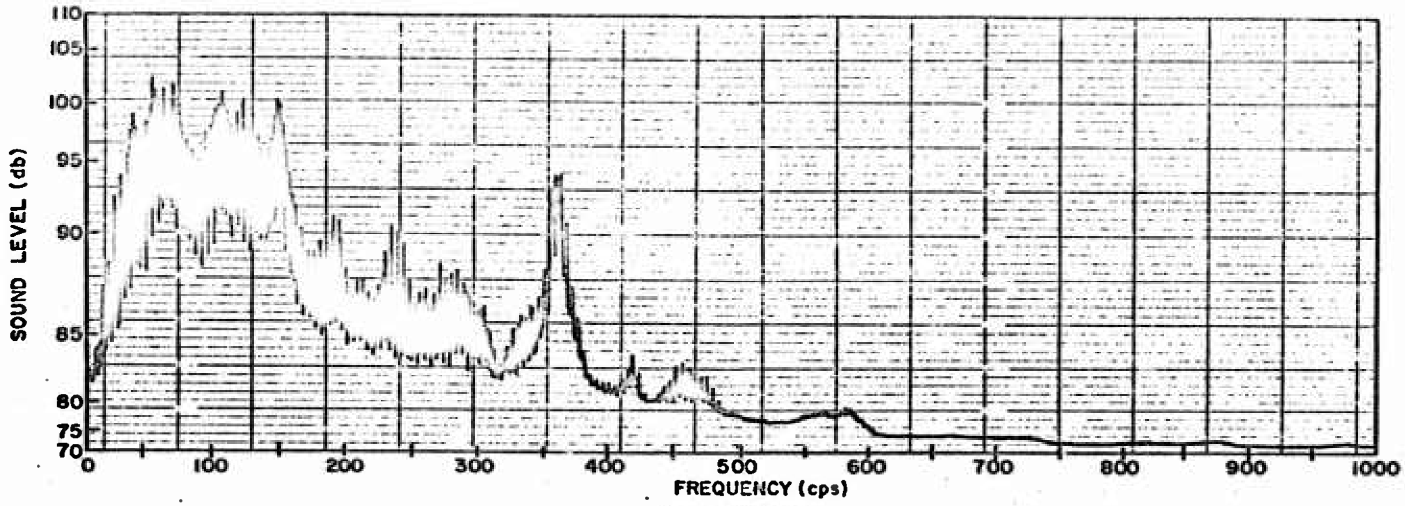


FIGURE 52

INTERNAL SOUND SPECTRUM, TOWING VEHICLE WITH THE TRACKS REMOVED AT 20 MPH, VEHICLE ENGINE SPEED OF 2400 RPM

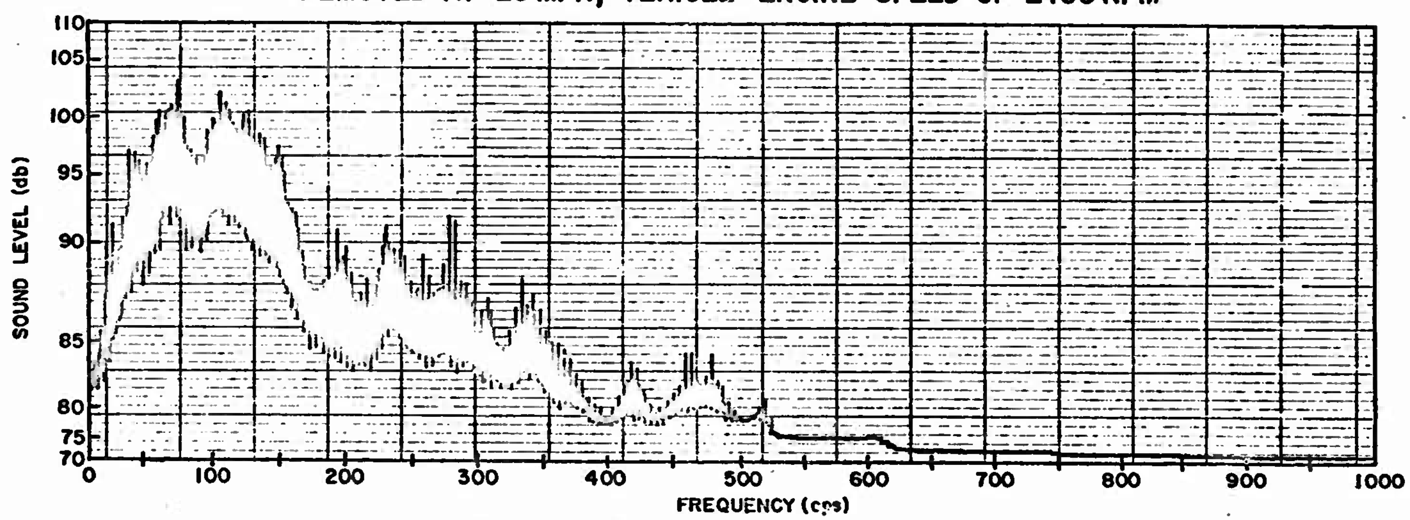


FIGURE 53

INTERNAL SOUND SPECTRUM, TOWING VEHICLE WITH THE TRACKS REMOVED AT 20 MPH, VEHICLE ENGINE OFF

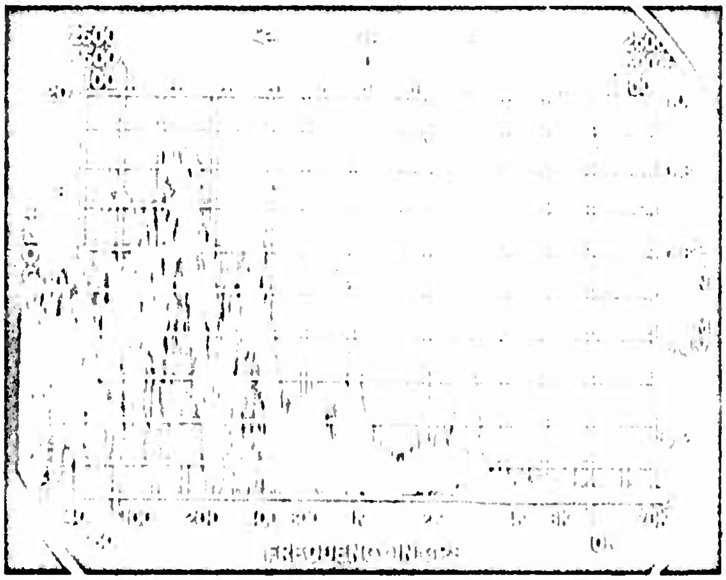


FIGURE 54. INTERNAL SOUND SPECTRA, IDLING ENGINE AT 2300 RPM. FULL SCALE DEFLECTION REPRESENTS 100 DB.

sprockets, idler wheels, and road wheels. Thus, testing was conducted in an attempt to evaluate the relative importance of the vibration transmission from these three elements.

a. Vibration Transmission From the Road Wheels

Comparing data taken towing the vehicle with and without the tracks gave considerable information on the importance of the track-road wheel reaction relative to track reactions with the sprockets and idlers. In addition to measuring internal sound level, transducers were mounted directly on road wheel support arms and torsion bars. Thus, it was possible to make direct correlation between suspension system vibrations and radiated internal noise. The results as given in Table XII indicated that although removing the track produced substantial reductions in the noise level, corresponding reductions in suspension system vibrations were not noted. In fact, comparing towing with the tracks on to towing with the tracks off on rough pavement, considerably higher vibrational velocities were recorded with the tracks off, but the sound level was 10 to 15 db less. Thus, it is indicated that the track-road wheel reaction is not as important a vibration source as the other track reactions. In Table XII the numbering of the road wheels and torsion bars start at the front of the vehicle, thus, measurements were made on the fourth and fifth road wheels from the front on the right side and on the rear two torsion bars. The velocites given indicate considerable fluctuations because of road surface variations.

Also, comparison of the vibration levels of the fourth and fifth road wheels illustrates the damping afforded by the use of shock absorbers since absorbers are mounted on the rear set of wheels. The transducers were mounted on the two support arms at approximately the same distance from the pivot point and the same orientation. The two support arms have dissimiliar shapes since the rear one has a shock absorber mounted on it, thus, making it impossible to record vibrations at identical positions on the two wheels.

In addition, considerable impulse testing data was taken to compare the relative vibration transmission from the sprocket wheels, idler wheels, and road wheels into the vehicle. The vibrations of various panels inside were measured with the CEC oscillograph when hammer blows were delivered to the wheels. In general, it was found that the response was less when striking the road wheels than when striking the sprockets or idlers. This is illustrated in Figure 55. In addition to measuring the vibrations, the internal impact noise levels were also recorded using the GR Impact Noise Analyzer in conjunction with the sound level meter. Several readings were taken striking each element. The average peak values obtained were 121 db

TABLE XII. SUSPENSION SYSTEM VIBRATION LEVELS COMPARING VARIOUS EFFECTS

Test Condition And Speed	RMS Vibrational Velocities, cm/sec R. Road Wheel Nbr. 4	R. Road Wheel Nbr. 5	Torsion Bar Nbr. 9	Torsion Bar Nbr. 10	Internal Noise Level db
Tracks On, 10 MPH	19-21	4-7	6-7	6-7	112-114
Tracks On, 20 MPH	15-70	6-9	12-14	13-36	118
Tracks Off, 10 MPH Rough Pavement	28-40	6-10	9-12	7-12	100
Tracks Off, 20 MPH Rough Pavement	59-70	21	30-46	27-45	108
Tracks Off, 10 MPH Smooth Pavement	5-6	4	5	3-4	96
Tracks Off, 20 MPH Smooth Pavement	8-12	5	13-16	14-21	104

TABLE XIII

VIBRATION MEASUREMENTS TRACING IMPULSES FROM ROAD WHEELS
(Values given are rms velocities in cm/sec)

Location of Transducer	Striking Right Road Wheel Nbr. 4	Striking Left Road Wheel Nbr. 4
Frame near left road wheel number 4 support mount	.5	.8
Frame near right road wheel number 4 support mount	1.1	.5
Left track fender	.75	1.45
Right track fender	1.8	.85
Left armor wall	.5	.65
Right armor wall	.65	.6

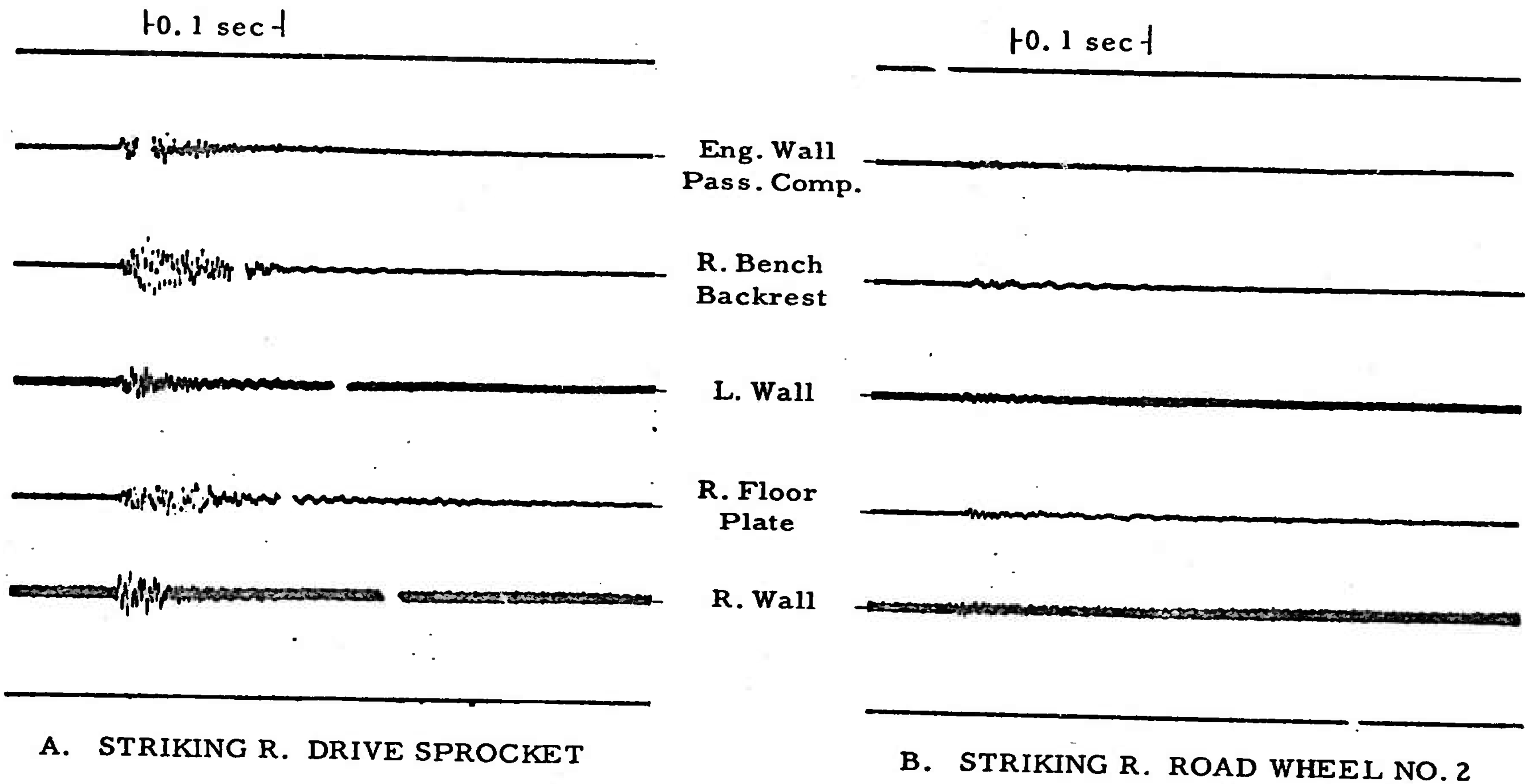


FIGURE 55. IMPULSE TESTS, COMPARING RELATIVE IMPEDANCES

when striking the drive sprockets, 119 db for the idler wheels, and 114 db for the road wheels. It should be realized that the impact data gives the response to impulses of comparable forces, and not to the actual forces involved while operating the vehicle.

Additional impact data was taken on the road wheels for the purpose of tracing the vibration transmitted from them to the vehicle through the two possible channels that are available (ie, either directly through the support bearing or through the torsion bar to the other side of the vehicle). The procedure used was to deliver an impact to a road wheel and measure the vibration at identical positions on both sides of the vehicle relative to this road wheel and its connecting torsion bar. The fourth set of road wheels from the front were used for the tests. Vibrations were recorded on the frame near where this set of road wheels and torsion bars are mounted and on the armor walls and track fenders directly over these wheels. The results indicated that there was a greater transmission of the initial impulse directly through the support bearing than through the torsion bar as shown in Table XIII.

b. Drive Sprocket and Idler Wheel Modifications

From our process of vibration source elimination, the track-sprocket and track-idler reactions are left as the worst offenders. It was shown that the track-idler reaction is important by the tests comparing the rubber and steel idlers. If it were possible to remove either the idlers or the sprockets and make tests with all the other conditions the same, it would be a simple matter to qualitatively compare the two reactions. It was possible to make tests with the idler wheels removed, but this created various complications in operating conditions. It was necessary to remove the tension cylinders, rear shock absorbers, and a sufficient number of track shoes so that the tracks enclosed the sprockets and road wheels only as shown in Figure 56. However, there was no way to adjust track tension properly and the importance of track tension on internal noise levels has already been demonstrated. Thus, the data taken with the idlers removed was compared to another set of full track data when track tension adjusted as nearly as possible to the non-idler condition. The tensions were compared by measuring the distance between the top of the front road wheel and the track. The results of these tests are given in Figure 57, indicating a reduction in noise at low speed without the idlers but a slight increase at high speed. However, it is difficult to draw firm conclusions from the tests because of the many variables involved.

Another method used for studying the track-sprocket reaction and the track idler reaction was to compare the relative effects

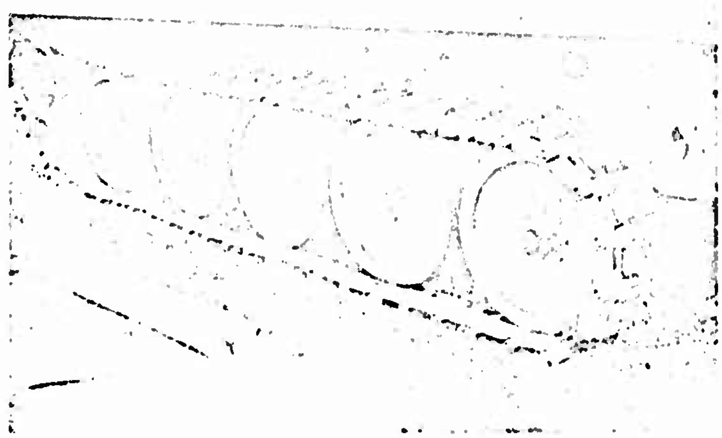
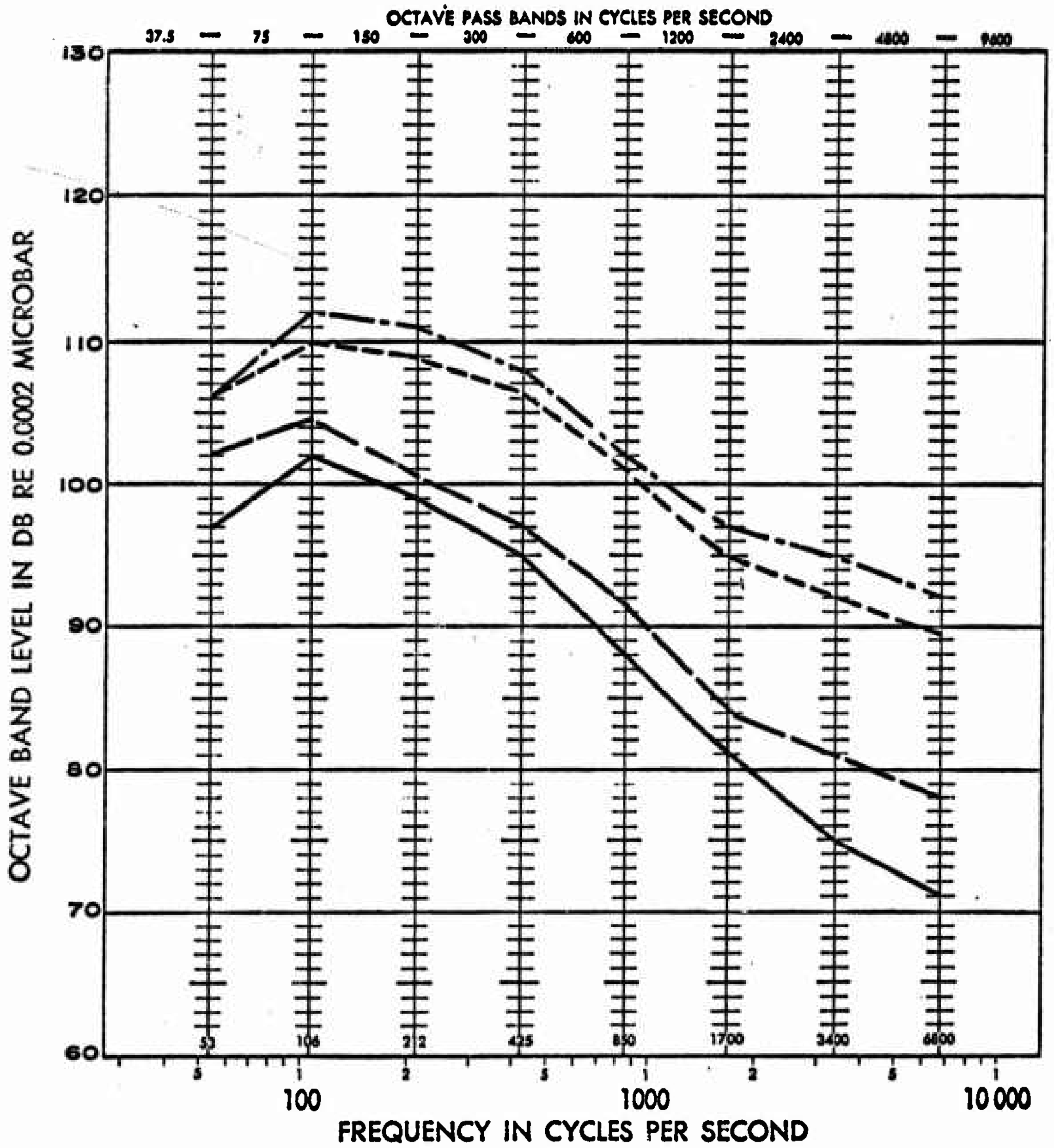


FIGURE 56. ILLUSTRATION OF TRACK CONFIGURATION WITHOUT IDLER WHEEL



10 MPH, IDLER WHEELS REMOVED
10 MPH, RUBBER IDLERS
25 MPH, IDLER WHEELS REMOVED
25 MPH, RUBBER IDLERS

FIGURE 57. INTERNAL NOISE SPECTRA SHOWING EFFECT OF REMOVING IDLERS.

produced by making various modifications of the sprockets and idlers. It was also envisioned that a simple modification of the sprocket and/or idler might lead directly to a method for substantially reducing noise levels. For all the following modifications tests, the vehicle was blocked up so that the tracks were free to turn without contacting the ground. It was more convenient for observing internal and external effects with the vehicle on blocks. Measurements were made at the standard internal position (ie, center of passenger compartment) and at external positions near the right drive sprocket and near the right idler. For the external measurements the microphone was held pointing downward in the vertical plane of the right side of the vehicle approximately 6" above the top of the sprocket or idler. Measurements were made at indicated speeds of 10 mph and 25 mph.

(1) Rubber Pads On Sprocket Teeth

It was felt that the sprocket was a worse offender than the idler because of the direct metal-to-metal contact between the sprocket teeth and the track shoes. Thus, first efforts were directed at eliminating this effect. This was accomplished by gluing a 3/16" thick strip of rubber elastomer (1/2" wide) directly on the sprocket teeth with contact cement. A diagram of the drive sprocket components is given in Figure 58. Data was taken while operating both in forward and reverse before the pads wore out. Octave band analyses of this data compared to the data operating without the rubber pads is given in Figures 59-61. The only noticeable improvement was in the high frequency bands measured outside near the right drive sprocket. In fact, the inside tests indicate that the overall noise level was higher with the pads than without at 25 mph. The major differences were in the lowest band and above 2000 cps.

(2) Rubber Belted Sprocket (Steel Teeth Removed)

For the next test the sprocket wheels (toothed parts) and the decagon shaped rubber sprocket cushions were removed, and round rubber tires consisting of layers of Conservo transmission belting were built up on the sprocket wheel hubs. The layers of the belting were glued together with contact cement. The total built-up thickness of the vulcanized rubber was about 1 1/8". The

(1) ILLUST		(2) SOURCE, MAINT AND RECOVERABILITY CODE				(3) FEDERAL STOCK NO	(4) DESCRIPTION	(5) UNIT OF ISSUE	(6) QTY INC IN UNIT	(7) 15-DAY MAINT ALW PER 100 EQUIP	
(a) FIG NO	(b) ITEM NO	(a) TECHNICAL SERVICE NO	(b) SOURCE	(c) MAINTENANCE LEVEL	(d) RECOVERABILITY					(a) CO OR BTRY	(b) SEP CO SEP BN OR REGT
							TRACK AND SUSPENSION - Continued				
							DRIVE SPROCKETS (Fig. 90)				
90	1		P	O		5305-660-2429	SCREW, CAP, HEXAGON HEAD: med-carb-S, cd- or zn-pltd, 1/2-20UNF-2A x 1 (96906-35304-109)	ea	40		
90	2		P	O	8	2520-679-7956	SPROCKET WHEEL: track drive (8763353)------------------------	ea	4		
90	3		P	O		2520-679-9657	CUSHION, RUBBER, sprocket (8763180)------------------------	ea	4		
90	4		P	O		5305-022-3808	SCREW, CAP, HEXAGON HEAD: alloy-S, cd- or zn-pltd, 5/8-18UNF-2A x 1-1/4 (96906-35304-160)	ea	20		
90	5		P	O	8	2520-679-7960	WHEEL, SPROCKET, final drive (8763352)------------------------	ea	2		

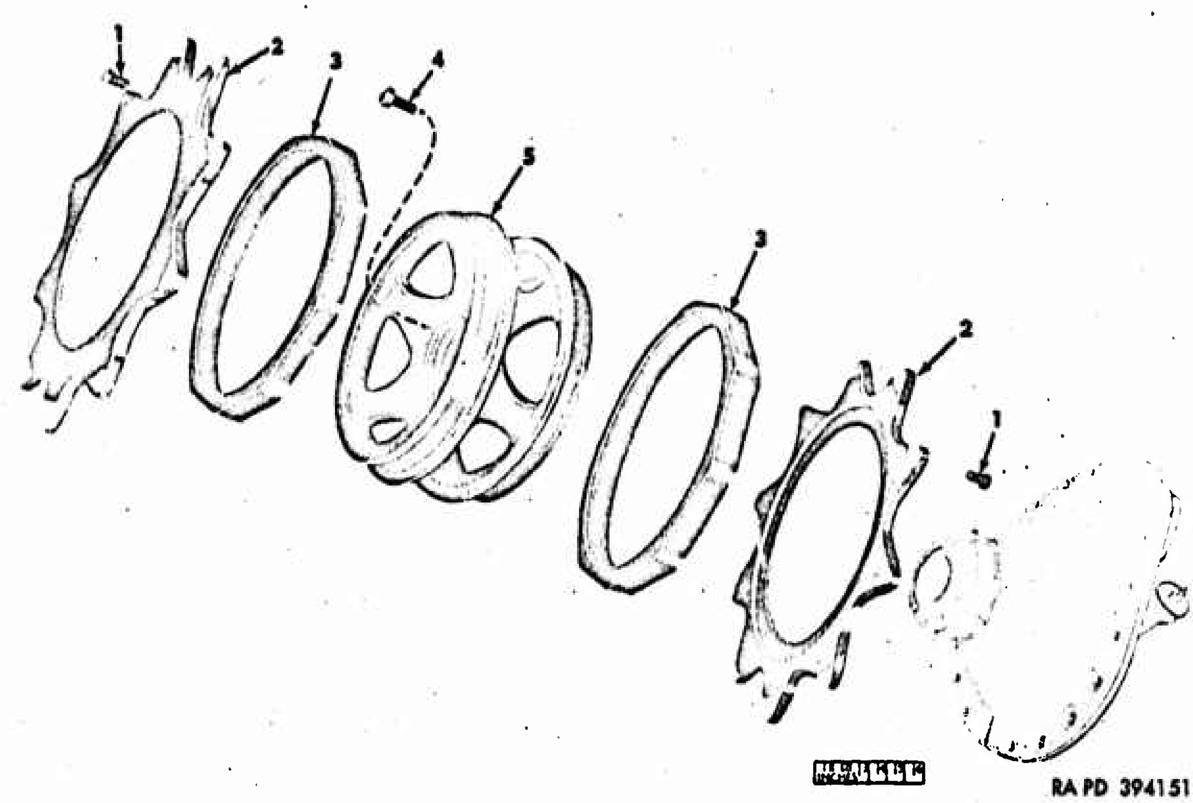
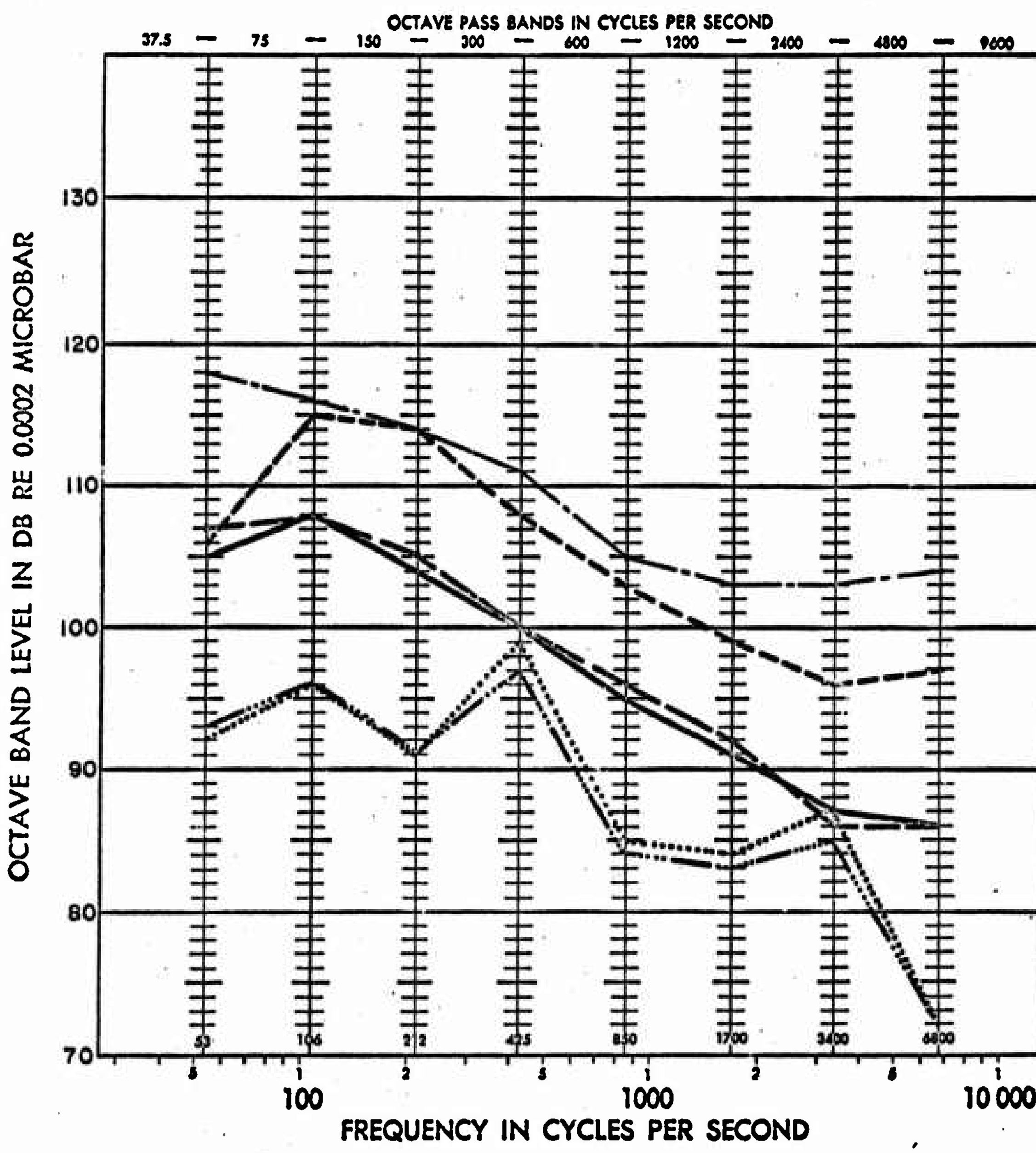


FIGURE 90–DRIVE SPROCKET–PARTIAL EXPLODED VIEW.

FIGURE 58. REPRODUCTION OF FIGURE 90 OF ARMY TECHNICAL MANUAL TM9-2300-224-20P



LEGEND:

- 10 MPH-NORMAL SPROCKETS.
- 10 MPH-WITH PADS ON SPROCKETS.
- 25 MPH-NORMAL SPROCKETS.
- 25 MPH-WITH PADS ON SPROCKETS.
- 2,000 RPM REVERSE NORMAL SPROCKETS.
- 2,000 RPM REVERSE WITH PADS ON SOCKETS.

FIGURE 59. INTERNAL SOUND SPECTRA WITH RUBBER PADS ON SPROCKET TEETH, VEHICLE ON BLOCKS.

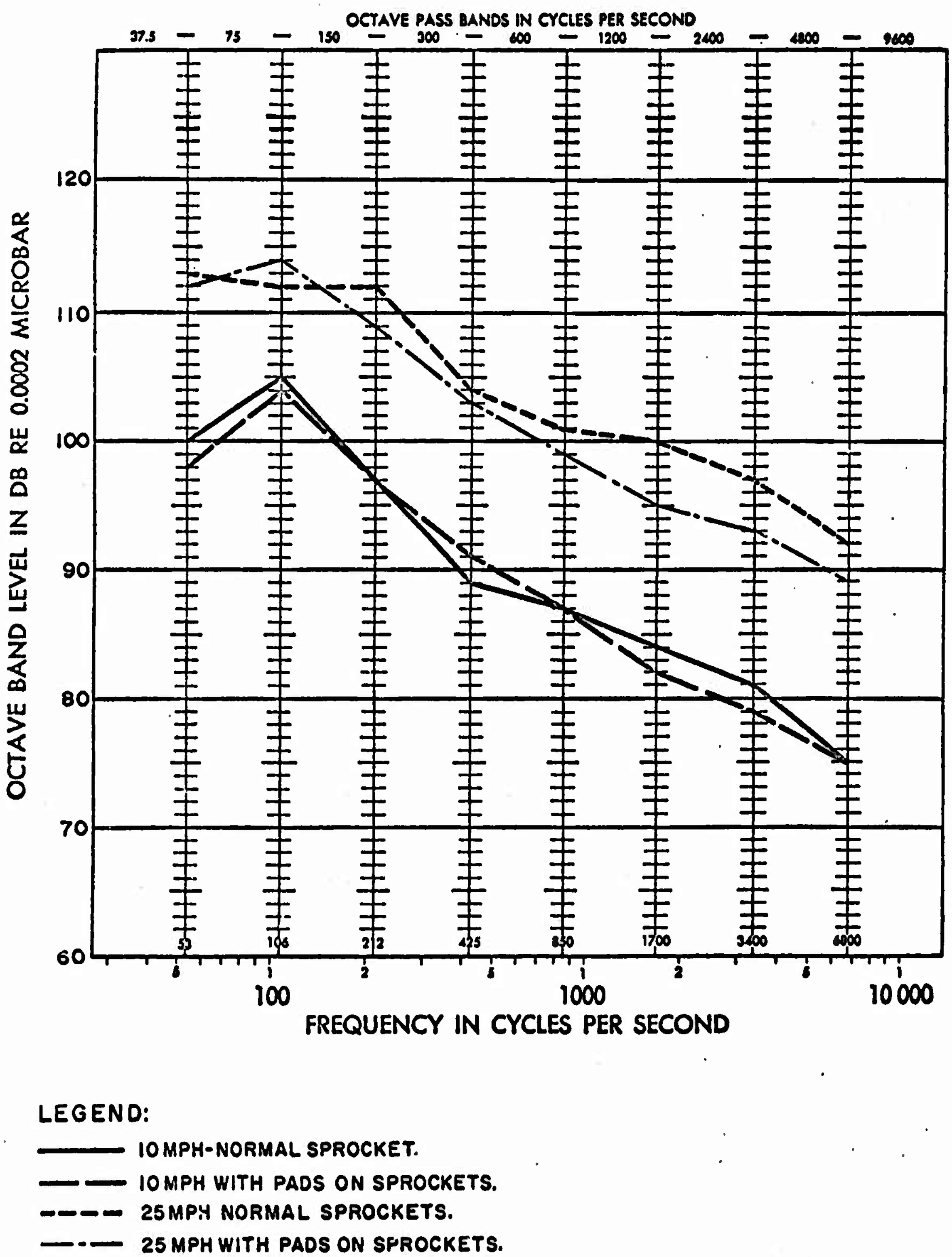


FIGURE 60. EXTERNAL SOUND SPECTRA WITH RUBBER PADS ON SPROCKET TEETH, MEASURED NEAR RIGHT SPROCKET, VEHICLE ON BLOCKS.

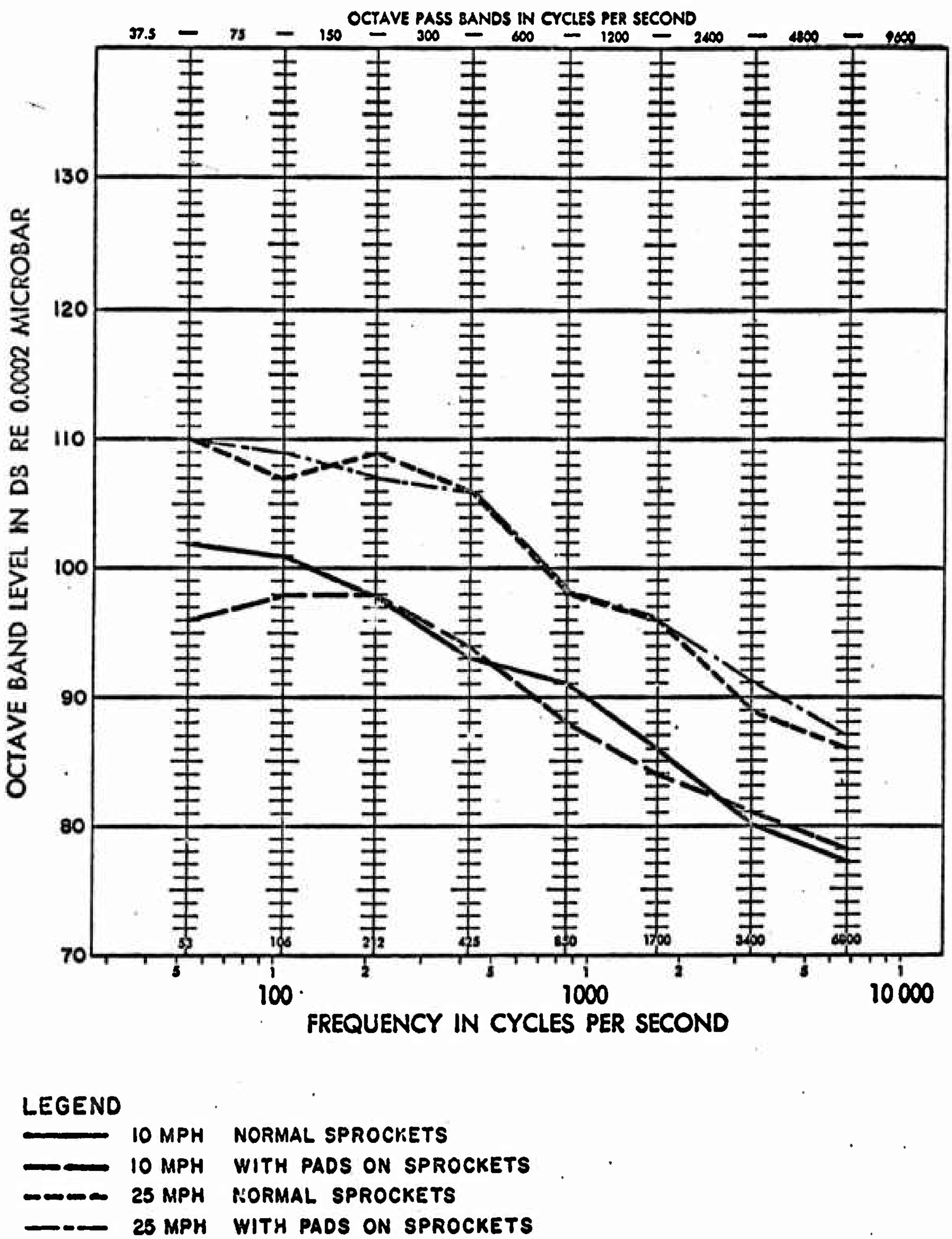


FIGURE 61. EXTERNAL SOUND SPECTRA WITH RUBBER PADS ON SPROCKET TEETH, MEASURED NEAR RIGHT IDLER WHEEL, VEHICLE ON BLOCKS.

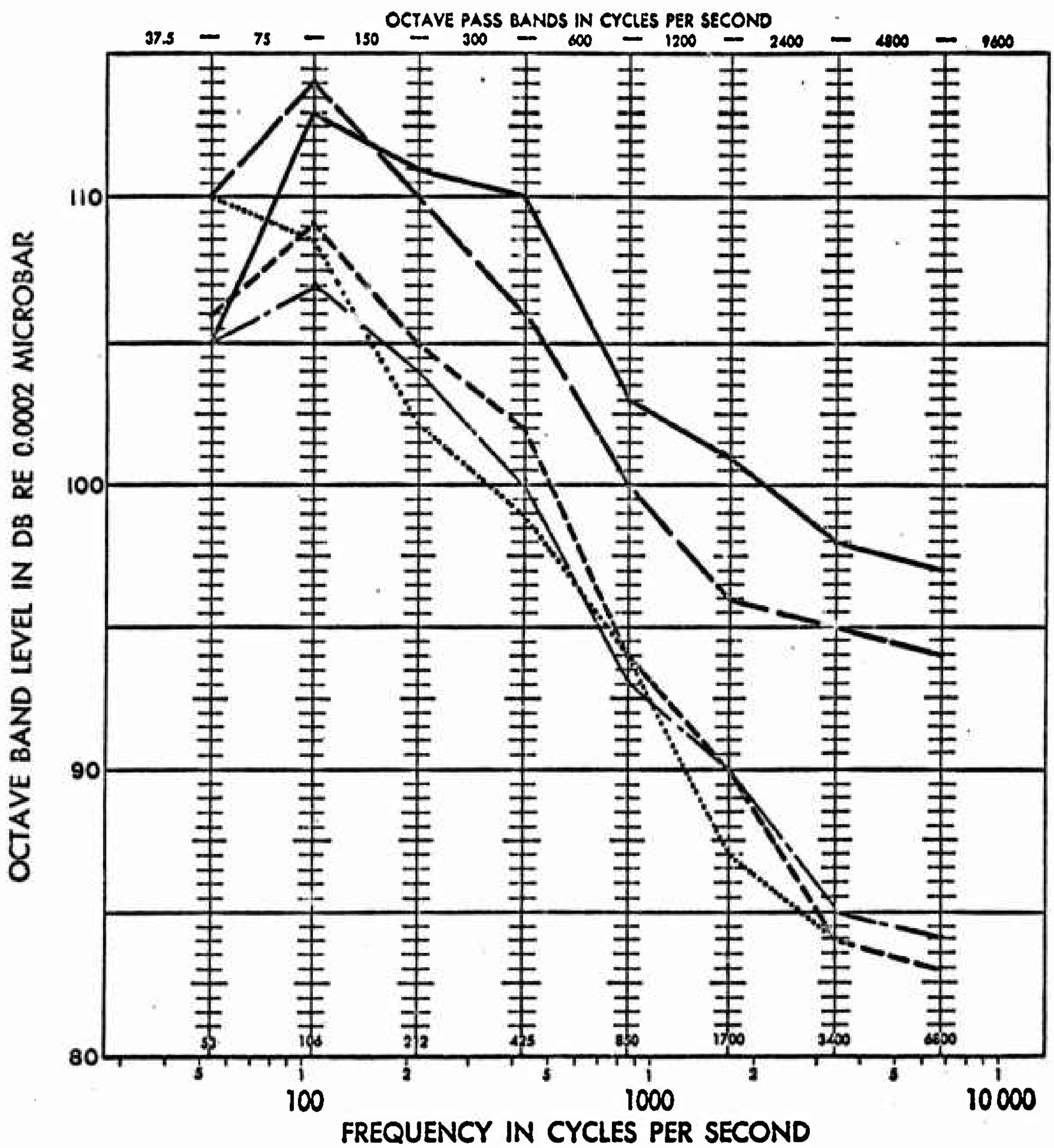
testing was limited to inside noise measurements due to the inability to guide the track without the sprocket teeth (i. e., the tracks would shift toward the vehicle and strike the hull). The results are compared to data taken with the normal sprocket wheels at the same track tension in Figure 62 and 63. These analyses indicate that the steel sprocket teeth contribute very little to the inside noise level, since the levels with and without them are nearly the same.

(3) Spacer Inserts in the Idler Wheels

Next, testing was conducted to evaluate the noise and vibration created by the striking of the track teeth on the inside surfaces of the idler wheels, which was noticed to be present during previous testing. To eliminate this effect, washers were inserted on the hub wheel bolts between the two halves of the idler wheels; thereby, the outside part was effectively moved out by the thickness of the washers. Three 0. 156 inch washers were used on each idler wheel to allow considerable clearance for the passage of the track teeth. During the testing made with these spacers inserted there was no noticeable rubbing. Tests were conducted inside the vehicle and outside near the right idler wheel. Results are given in Figures 62-65. The outside noise measurements indicate a considerable contribution of high frequency by this "rubbing" effect at a speed of 25 mph. Inside noise measurements show little difference between tests with and without the spacers, indicating that this "rubbing" reaction is not a major vibration source.

(4) Rubber Idlers Compared to Steel Idlers

Tests were conducted to compare the rubber idlers to the steel idler wheels to supplement the vibration and sound data taken previously for this purpose. The results given in Figures 62-65 indicate that the steel idlers are definitely more noisy both inside and outside the vehicle especially at 10 mph which agrees with the data taken when operating on the ground.



LEGEND

- ——— RUBBER CUSHIONS REMOVED FROM SPROCKET WHEELS, STEEL IDLERS.
- — — NORMAL SPROCKET WHEELS, STEEL IDLERS.
- - - - NORMAL SPROCKET WHEELS, RUBBER IDLERS.
- — - — NORMAL SPROCKET WHEELS, RUBBER IDLERS WITH SPACERS INSERTED.
- ··········· RUBBER BELTED SPROCKETS (NO TEETH), RUBBER IDLERS.

FIGURE 62. INTERNAL SOUND SPECTRA AT 10 MPH WITH VARIOUS SPROCKET AND IDLER MODIFICATIONS, VEHICLE ON BLOCKS.

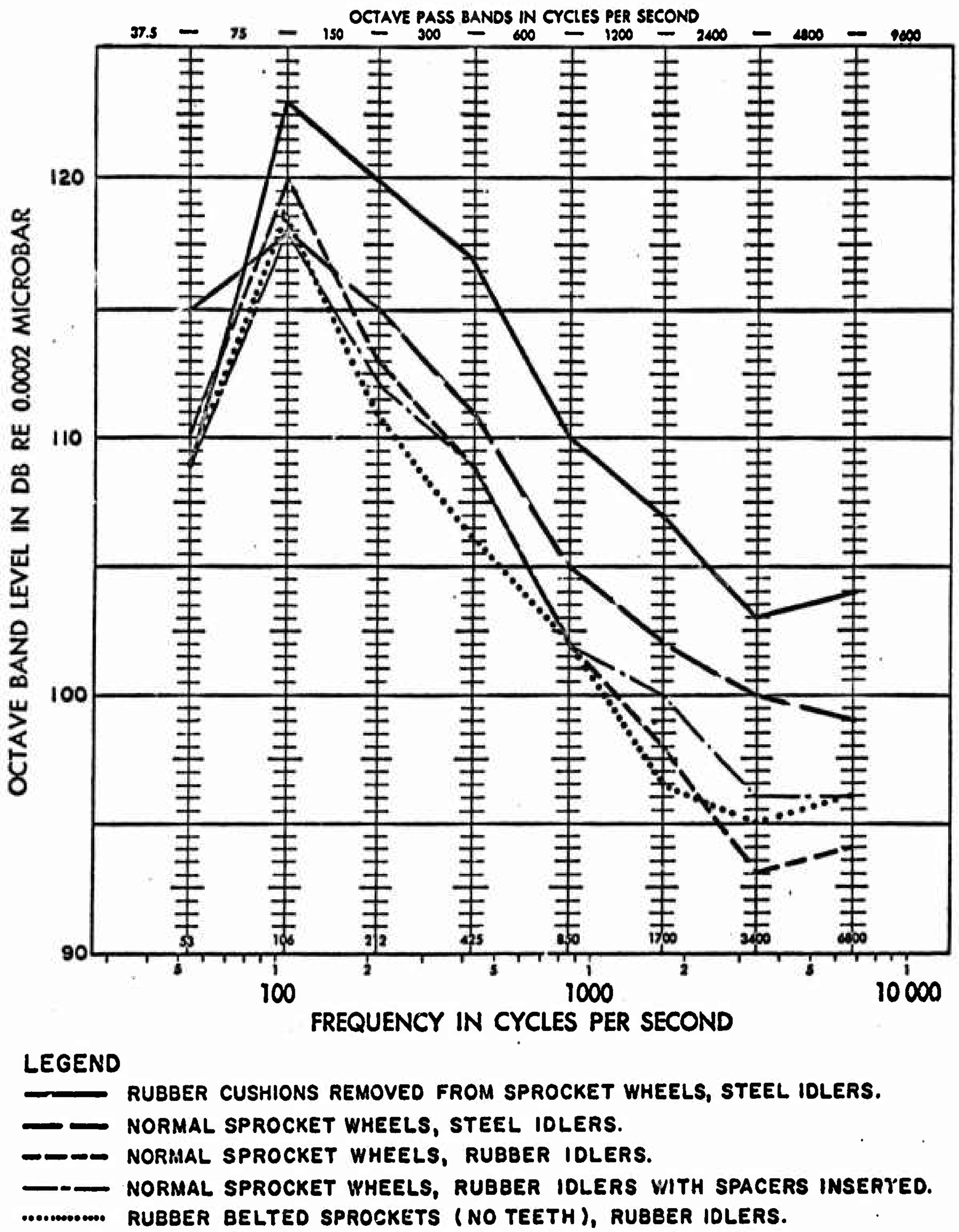
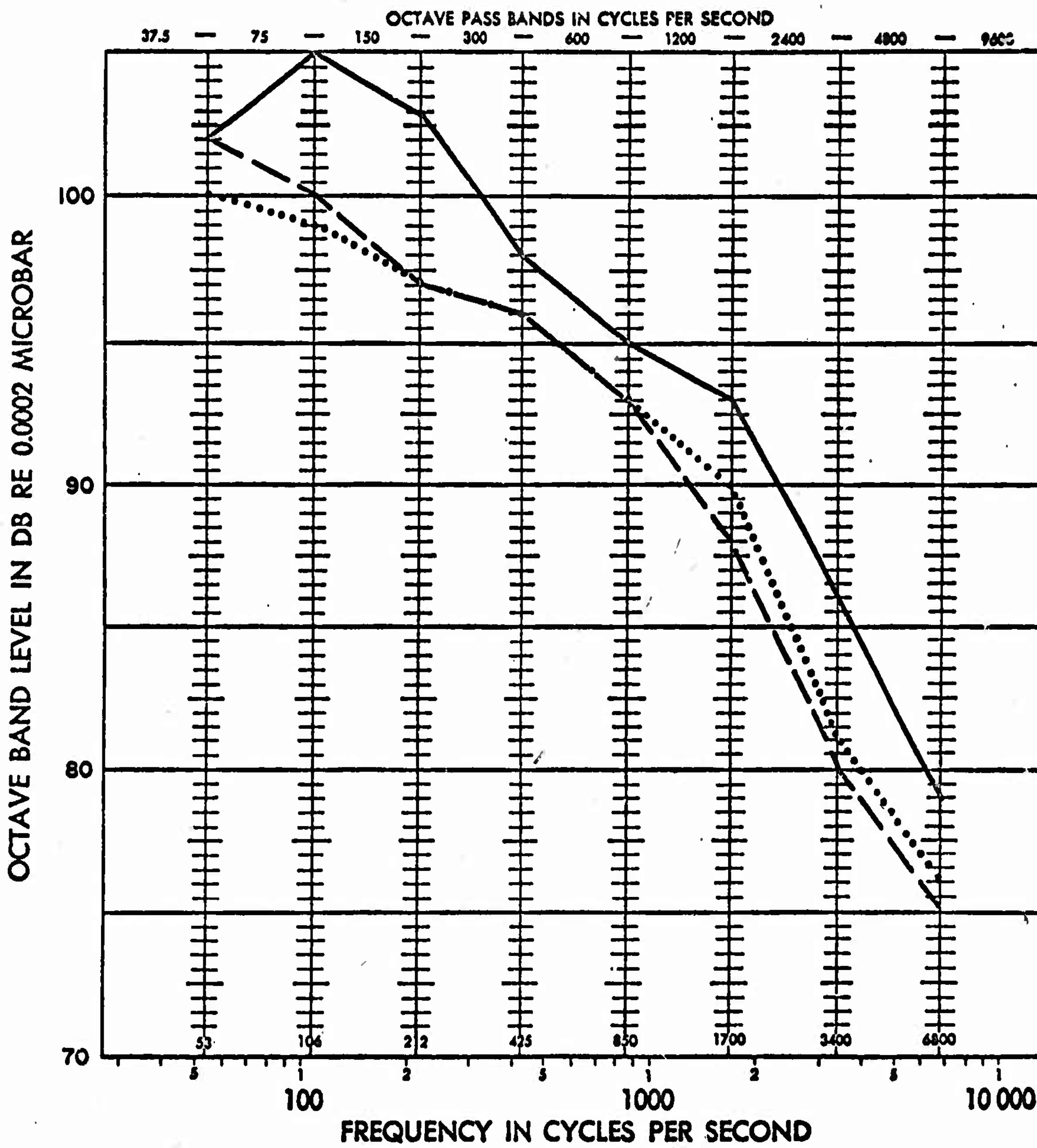


FIGURE 63. INTERNAL SOUND SPECTRA AT 25 MPH WITH VARIOUS SPROCKET AND IDLER MODIFICATIONS. VEHICLE ON BLOCKS.



LEGEND

——— STEEL IDLERS

.... RUBBER IDLERS

— — — RUBBER IDLERS WITH SPACERS INSERTED

FIGURE 64. EXTERNAL SOUND SPECTRA AT 10 MPH WITH VARIOUS IDLER MODIFICATIONS MEASURED NEAR RIGHT IDLER WHEEL, VEHICLE ON BLOCKS.

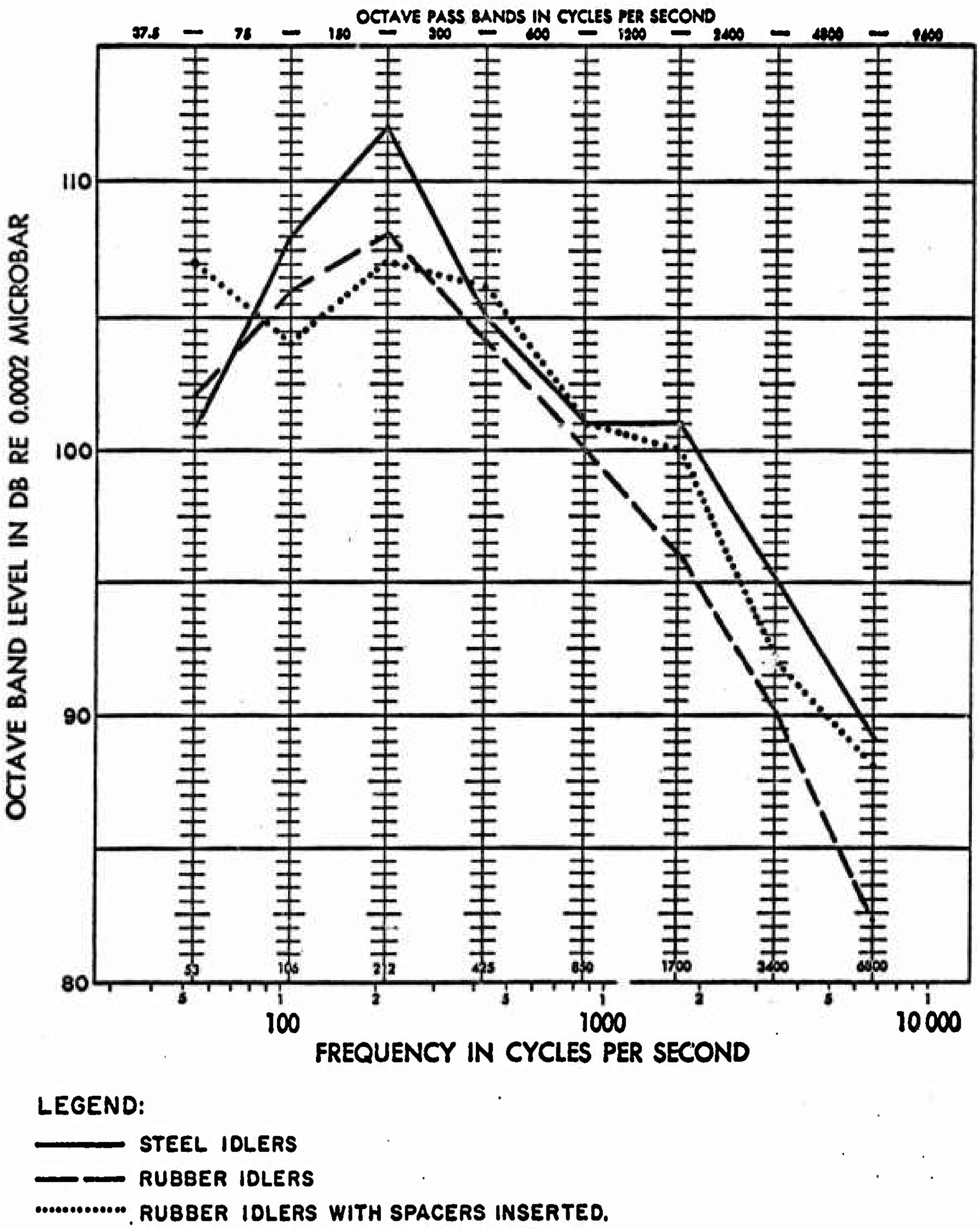
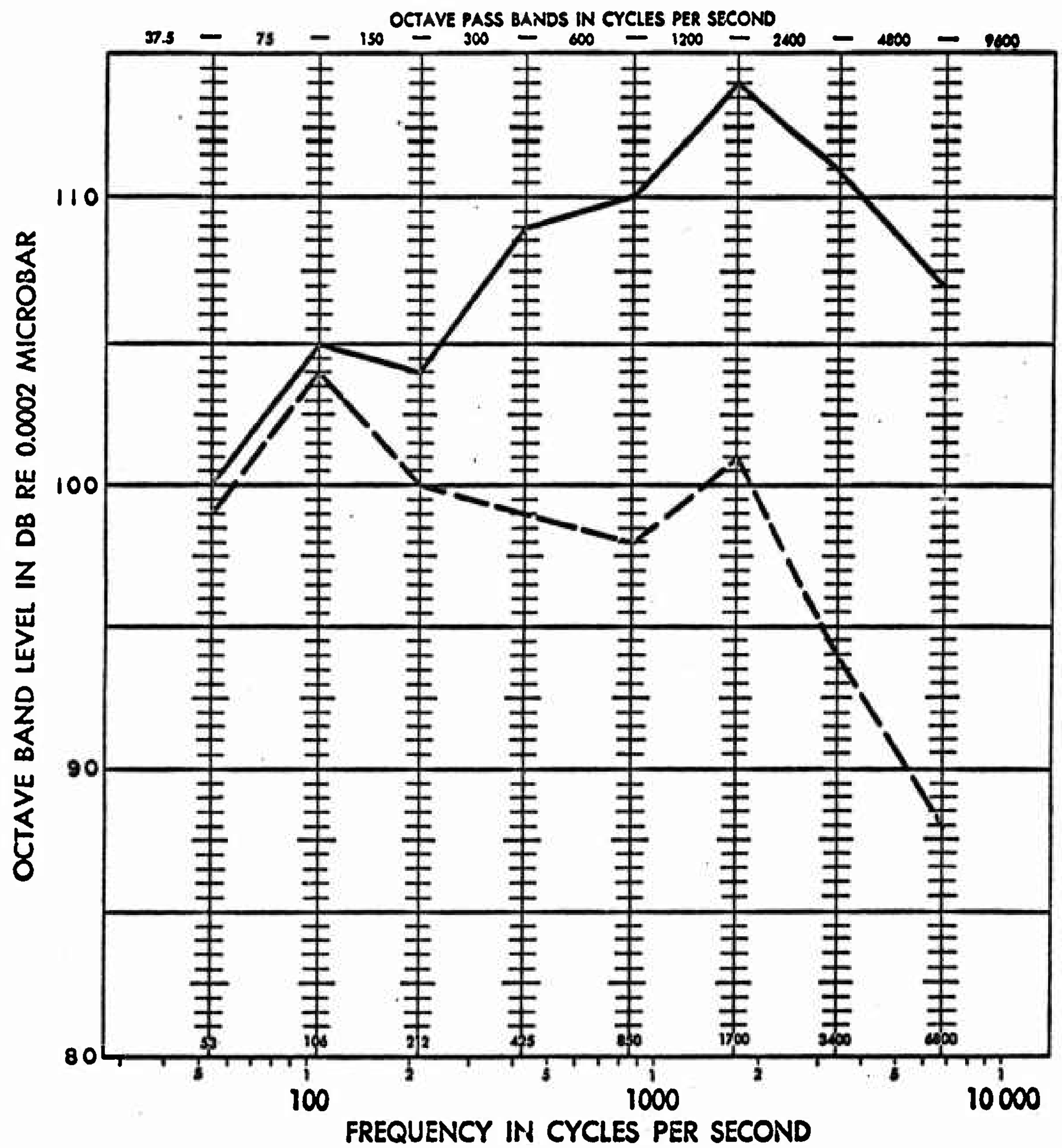


FIGURE 65. EXTERNAL SOUND SPECTRA AT 25 MPH WITH VARIOUS IDLER MODIFICATIONS MEASURED NEAR RIGHT IDLER WHEEL, VEHICLE ON BLOCKS.

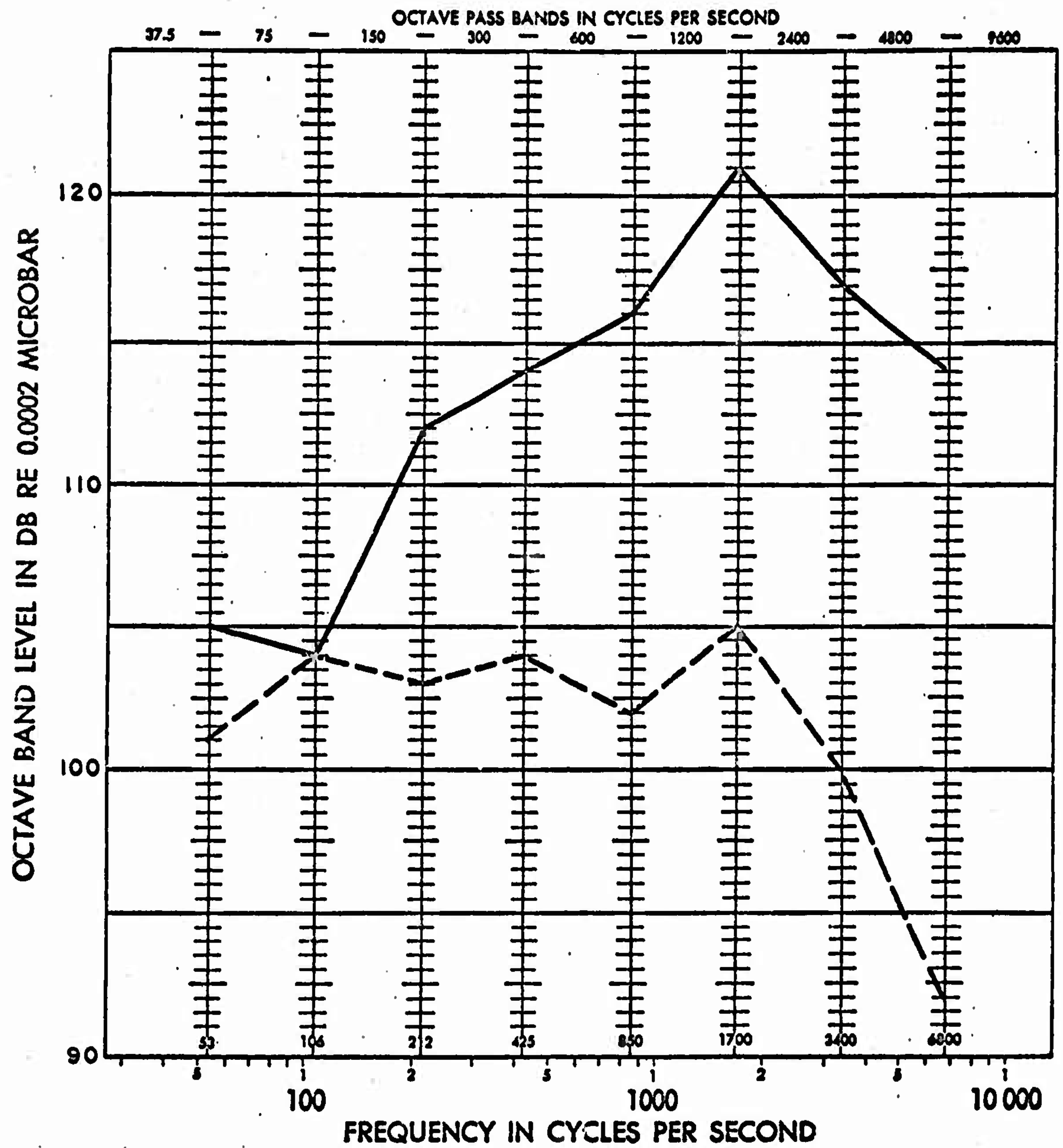


LEGEND

— — — NORMAL SPROCKET WHEELS

——— RUBBER CUSHIONS REMOVED FROM SPROCKET WHEELS

FIGURE 66. EXTERNAL SOUND SPECTRA AT 10 MPH MEASURED NEAR THE RIGHT SPROCKET WITH AND WITHOUT RUBBER CUSHIONS ON THE SPROCKETS, VEHICLE ON BLOCKS.



LEGEND

- - - - NORMAL SPROCKET WHEELS
——— RUBBER CUSHIONS REMOVED FROM SPROCKET WHEELS

FIGURE 67. EXTERNAL SOUND SPECTRA AT 25 MPH MEASURED NEAR THE RIGHT DRIVE SPROCKET WITH AND WITHOUT THE RUBBER CUSHIONS ON THE SPROCKETS, VEHICLE ON BLOCKS.

b. Tow-by at 10 mph and 20 mph with the vehicle engine running at an rpm comparable to that observed when driving at the same speeds.

c. Tow-by at 10 mph and 20 mph with the vehicle engine off.

d. Driving only the tow truck by at 10 mph and 20 mph to determine the noise it produces.

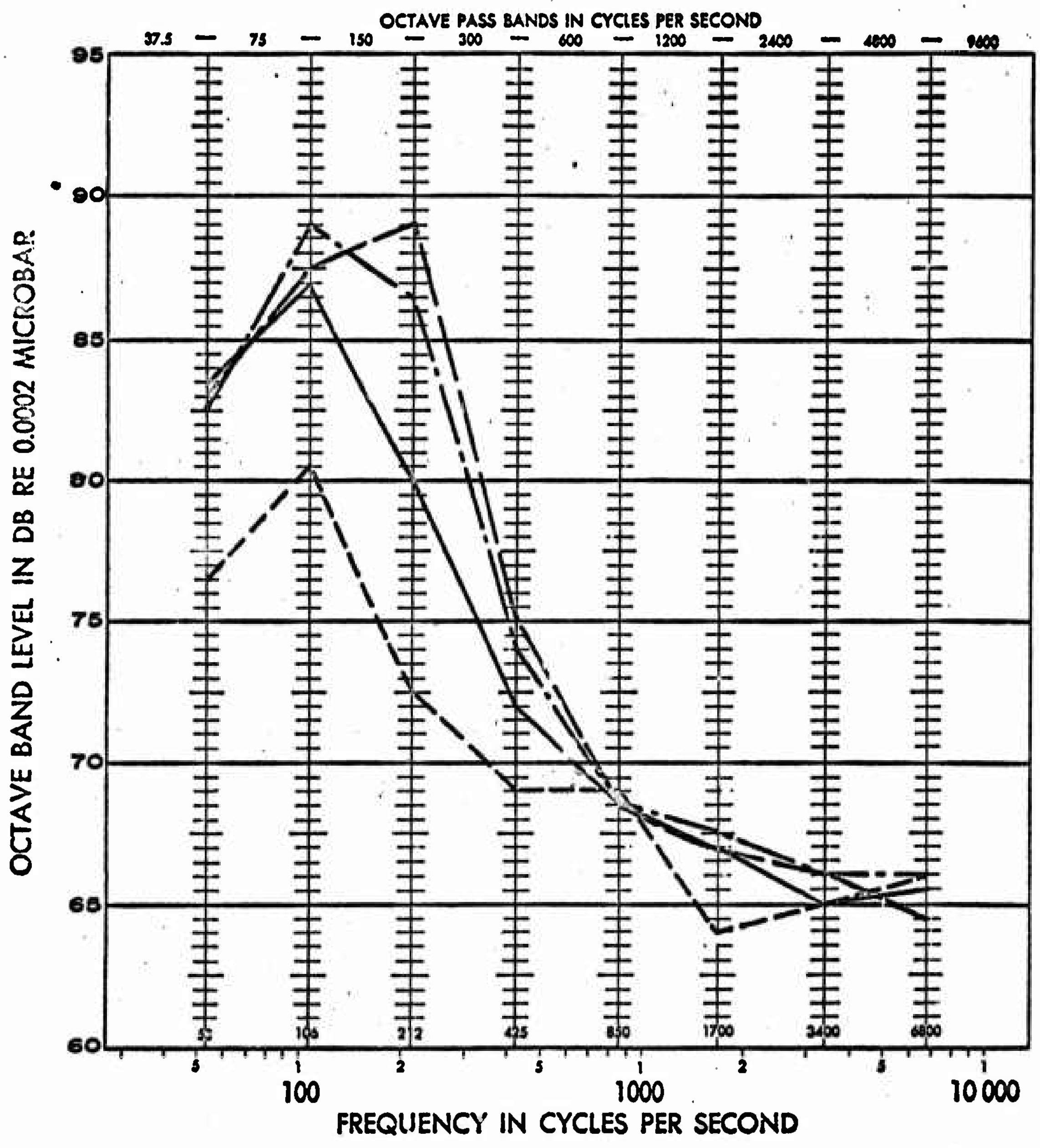
The results are presented in Figures 68 and 69. It is indicated that externally as well as internally the tracks are the major source of noise. Tow-by tests were attempted with the tracks removed but the levels observed were only slightly above the tow truck noise level except at high speeds.

E. Treatment Testing

Throughout the testing period the feasibilities of various simple treatment techniques were studied. These treatments included vibration source modification, noise source modification, and absorption techniques. Many of the techniques have been described previously in conjunction with vibration and noise source identification. Thus, in this section only the tests that were tried specifically for treatment purposes are described in detail, and a brief notation is made of the tests which served dual purposes.

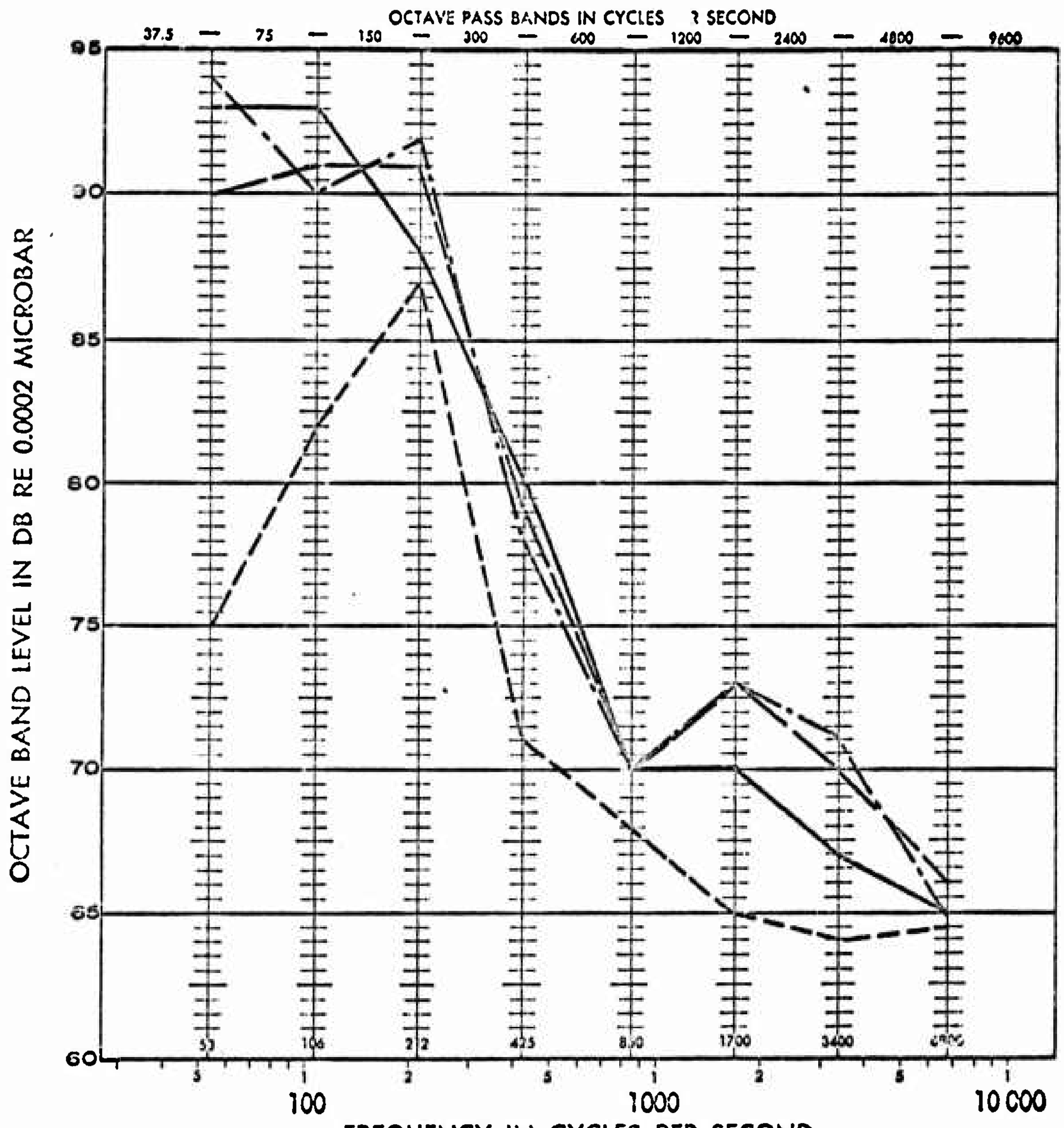
1. Panel Stiffening

The value of applying stiffening devices to panels with low damping rates was investigated. 2" x 1" aluminum channels, four feet long, were clamped centrally to the two bench back rests. Clamps were placed at the center of each channel and about 3" from either end. The effect on the fundamental frequency and natural damping rate of the left bench back rest is illustrated in Figure 70, comparing impulse tests on the back rest with and without the channel clamped on. The measurements were made near the center of the panel. The natural frequency was raised from 38 cps to 42 cps and the solid damping factor from .005 to .025. For the right back rest the natural frequency was raised from 39 cps to 45 cps and the solid damping factor from .039 to .055. In actual operation of the vehicle, a reduction by a factor of two was observed in the vibration of the left bench back rest as shown in the coasting test data in Table VII. Thus, it appears quite feasible that panel stiffening techniques or else using a different material with more inherent damping in place of the thin vehicle panels would produce considerable reductions in radiated noise levels.



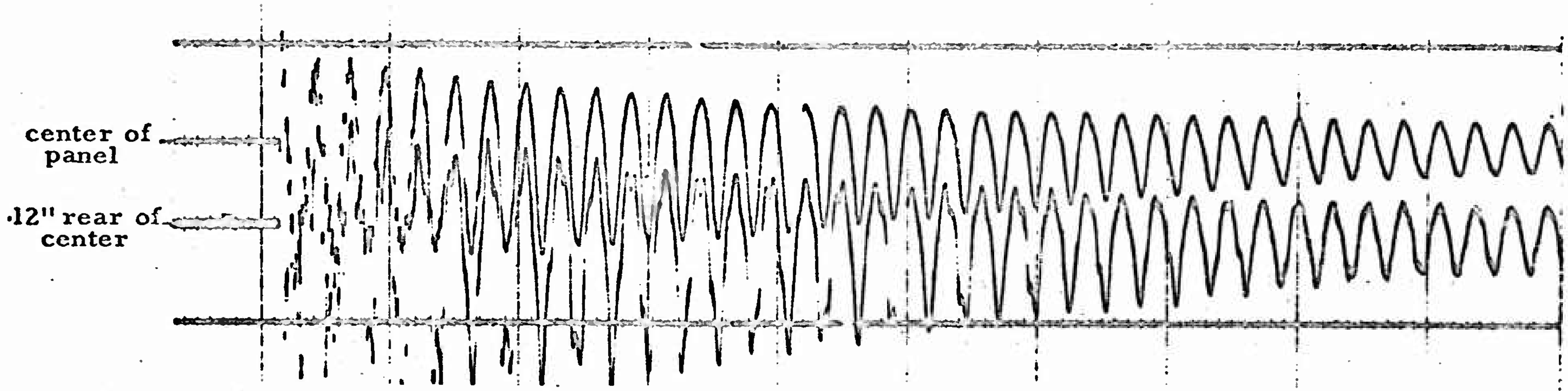
DRIVEBY
TOWBY, ENGINE IDLING
TOWBY, ENGINE OFF
TOW TRUCK NOISE

FIGURE 68. EXTERNAL NOISE SPECTRA COMPARING NOISE SOURCES, 10 MPH.

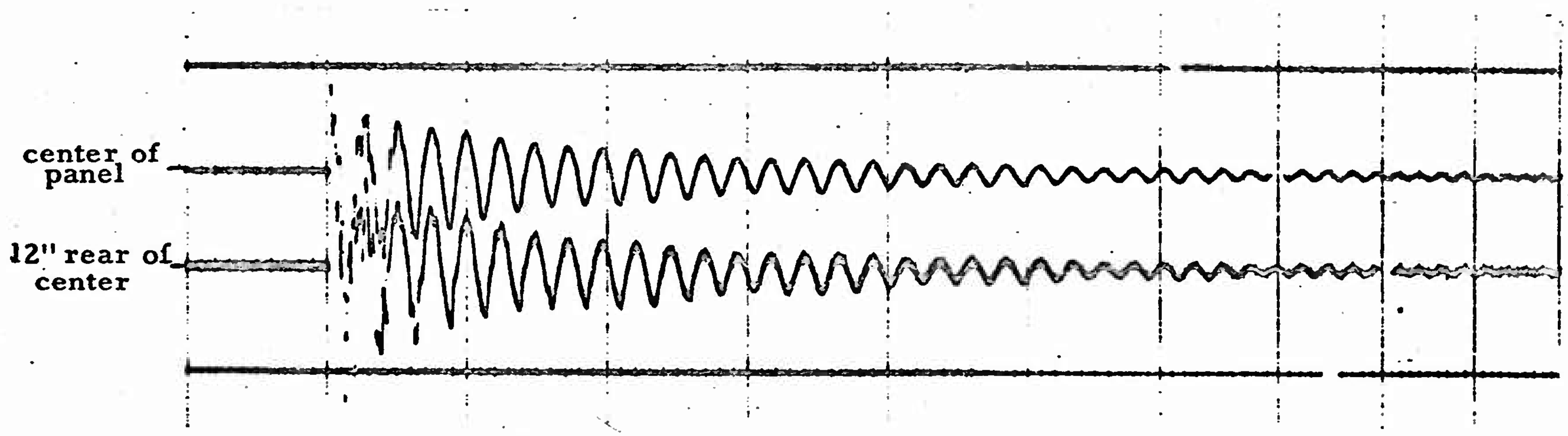


DRIVEBY
TOWBY, ENGINE IDLING
TOWBY, ENGINE OFF
TOW TRUCK NOISE

FIGURE 69. EXTERNAL NOISE SPECTRA COMPARING NOISE SOURCES, 20 MPH.



A. Left Bench Back Rest



B. Left Bench Back Rest with Channel

FIGURE 70. IMPULSE TESTS SHOWING EFFECT OF CHANNEL UPON VIBRATION OF BACK REST PANEL

2. Acoustic Absorptive Panel Testing

Tests were conducted to determine the possibilities of reducing the sound level inside the vehicle by adding sound absorptive material. The particular material used for the tests was a 2" layer of fiber glass attached to a 1/2" layer of plywood. For feasibility studies, panels of this material were cut identical in shape to the vehicle floor panels, and testing was conducted to evaluate the absorption provided by these panels. The testing consisted of measuring the internal noise level at speeds of 10 mph and 25 mph with the absorptive floor panels, then with the regular vehicle floor panels, and again without any floor panels. The testing was conducted on the regular test course. The absorptive panels were installed with the plywood side up. The results are given in Figures 71 and 72.

Also, the effect produced by adding more of the acoustic absorbing material was tested. These additional panels were not cut to fit in any particular position but were inserted along the walls of the passenger compartment in the most suitable manner. These panels were either strapped or hand held in place during the testing. The intention was merely to test the effects of additional absorption in the vehicle, not to consider in detail absolute fits or rigid mountings. Testing was done with the absorbing side toward the walls and again away from the walls. The approximate size of the panels and their respective positions were:

> A 4' x 3 1/2' panel on the rear ramp; a 2' x 6' panel along each side of the passenger compartment; and three panels 2' x 4' along the engine wall and access panel in the passenger compartment.

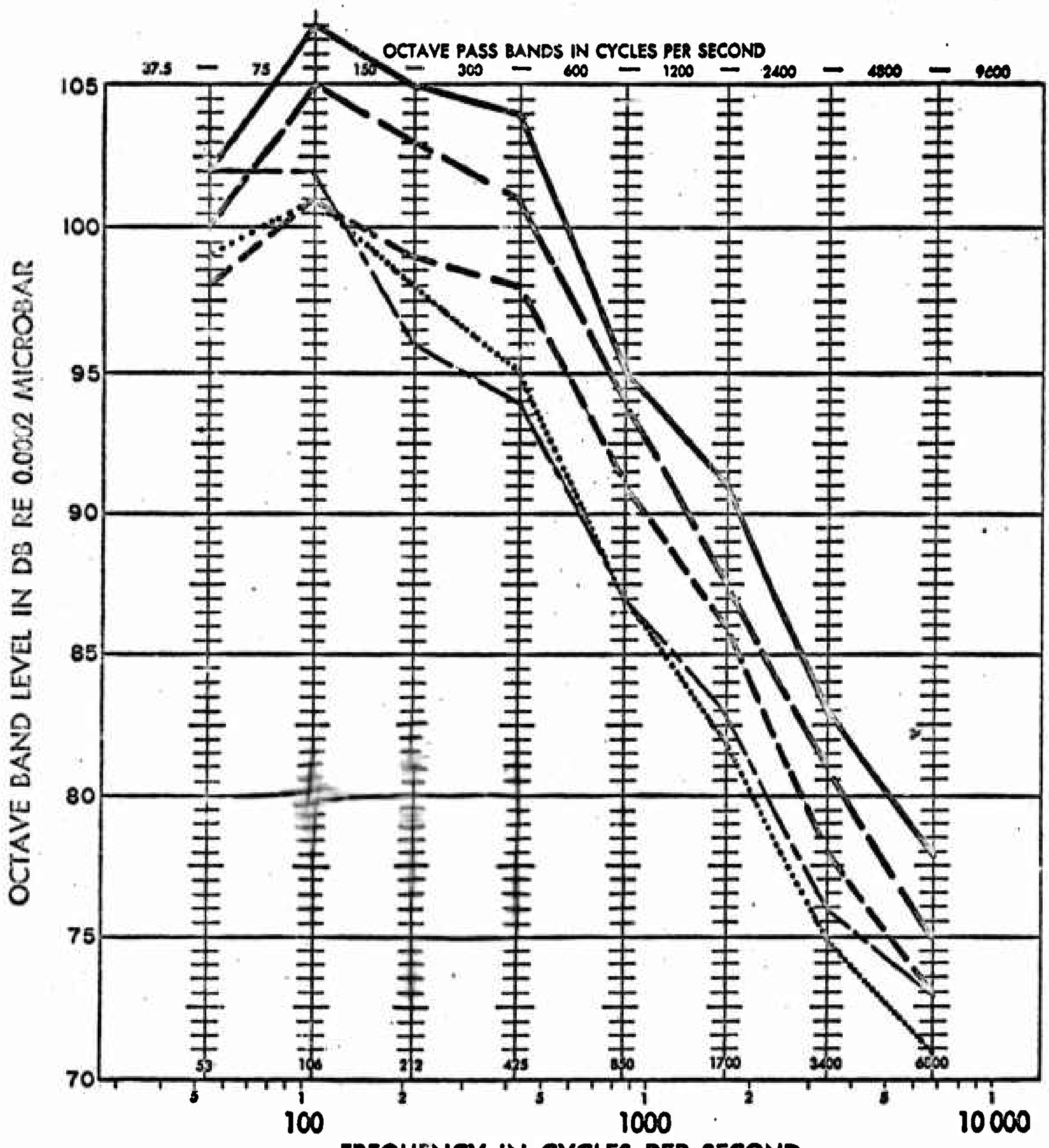
The results of the various tests as shown in Figures 71 and 72 look quite promising. A reduction of 5 db to 10 db is produced in each octave when using all of the acoustic paneling. The acoustic floor paneling produced their greatest effect at low frequency while the wall paneling produced substantial reductions at high frequency. Very little difference is evident between placing the wall panels with the acoustic side in or out.

3. Rubber Idler Testing

It has already been shown that replacing the steel idlers with rubber idlers produces a noticeable reduction in noise levels especially at low speeds.

4. External Muffler Effect

Likewise it was shown that the installation of an external



LEGEND

- REGULAR FLOOR PANELS
- NO FLOOR PANELS
- ACOUSTIC FLOOR PANELS
- ACOUSTIC FLOOR AND WALL PANELS (ABSORPTIVE SIDE TOWARD WALLS)
- ACOUSTIC FLOOR AND WALL PANELS (ABSORPTIVE SIDE AWAY FROM WALLS)

FIGURE 71. INTERNAL SOUND SPECTRA AT 10 MPH SHOWING EFFECTS OF ACOUSTIC ABSORPTIVE PANELING.

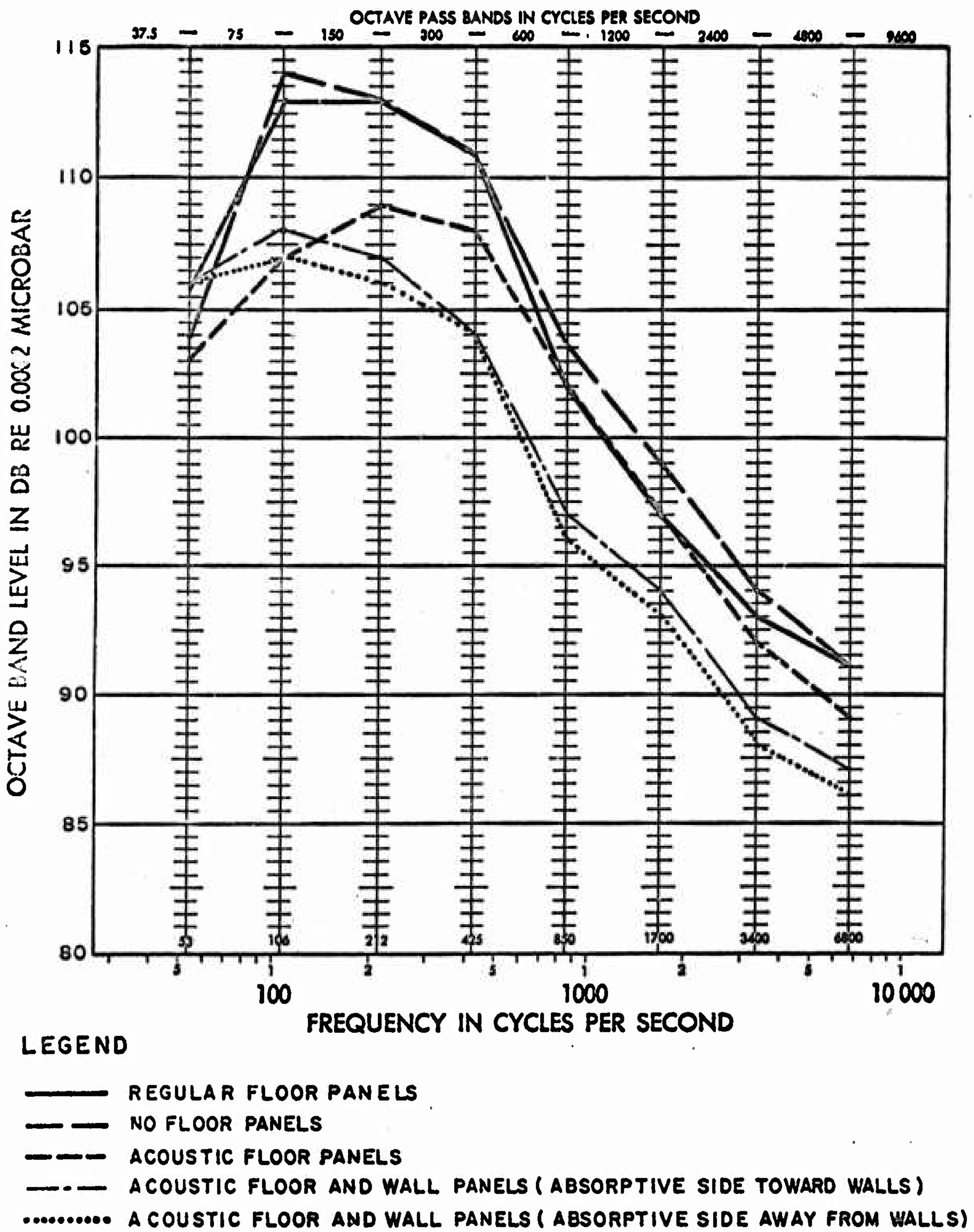


FIGURE 72. INTERNAL SOUND SPECTRA AT 25 MPH SHOWING EFFECTS OF ACOUSTIC ABSORPTIVE PANELING.

muffler produced substantial reduction in engine noise. However, until the track noise is reduced considerably, the effect of the muffler would be relatively insignificant except under idle conditions or at low speeds.

5. Other Vibration Source Modifications

As mentioned in a preceding section, several modifications were made of the sprockets and idlers. However, the results indicate, at best, only a minor improvement was made.

F. Summary of Noise and Vibration Testing

Through the use of the instrumentation systems that were developed in conjunction with the evolved systematic test sequence, it was possible to effectively evaluate the vibration and noise characteristics of the M-113 tracked vehicle.

It was found that very severe internal noise conditions exist in the vehicle while in operation. For vehicle speeds above 10 mph, the octave band noise levels are above the maximum acceptable level for Army Materiel Command equipment as specified by the Human Engineering Laboratories as of October 1963 with the exceptions of the lowest and highest octave bands. These intense internal sound levels are the resultant of the noise radiation from the numerous panels in the vehicle. In particular, the thin, low damped metallic panels such as the engine access panels, floor panels, bench back rests, etc., were shown to be potentially the worse noise offenders. The potentialities of the various noise radiators are summarized in Table VIII. The internal noise is characterized by strong low frequency spectra consisting predominantly of components corresponding to the major driving frequencies, namely those resulting from the track reactions, and of components attributable to the various panel fundamental frequencies. Additional complications arise from standing-wave patterns.

The noise radiators are set into vibration by energy transmitted to them from the various vibratory sources. The principals of these sources are the tracks and their reactions. The most important track reactions were found to be those with the drive sprockets and idler wheels with the track-road wheel reactions being of a secondary nature. Vibrations are transmitted from the sprockets and idlers through the hull of the vehicle to the noise radiating panels. Next in importance as vibration sources are the suspension system and then the engine. The relative importance of the vibration sources are stressed by the results of Table X.

Various treatment techniques tried consisted of vibration isolation, vibration damping, and sound absorption. Vibration isolation methods applied to the drive sprockets and idlers for the reduction of vibration at the sources showed some promise of success although more refined techniques are necessary for fully exploiting the possibilities. It was also demonstrated that the application of vibration damping techniques to the important noise radiating panels could effectively reduce their potential. Furthermore, the addition of good acoustic absorptive material to the inside of the vehicle gave quite favorable results. Thus, although the problems are severe and complicated, there exist numerous possible solutions.

IV. RECOMMENDATIONS

A. Recommended Measurement Techniques for the Analysis of the Vibration and Noise Characteristics of Tracked Vehicles

The principle objective of the overall program has been to develop measurement techniques that can be used to effectively identify and evaluate vibration and noise sources and the vibration transmission paths that exist in tracked vehicles. During a testing program conducted on an M-113 Armored Personnel Carrier, numerous instrumentation and measurement methods were formulated and evaluated for their relative effectiveness in accomplishing the desired goals. The basic techniques evolved as a result of this work should be applicable to almost any type of tracked military vehicle, although some adaptations may be necessary to apply the techniques to specific configurations and problems associated with other types of vehicles. However, if the basic underlying principles of the methods are understood, these adaptations should be relatively straightforward for most tracked vehicle noise problems. This section presents a synopsis of the evolved instrumentation and test techniques, and a condensation is given in Table XIV.

1. Instrumentation

There are several general requirements which can be specified for an instrumentation system to be used in sound and vibration studies for tracked vehicles. First, all measuring and recording systems must have a flat response throughout the frequency range of interest, which is essentially from 0-10,000 cps for vibration and 40-10,000 cps for sound. This requires that all system components such as transducers, microphones, amplifiers, recorders, and analyzers have flat frequency responses throughout these ranges.

The primary governing factor in the formulation of any instrumentation system, however, is the type of information that is desired. For example, one important requirement for any noise and vibration study is a method for making spectral analyses to the degree of detail necessary. For tracked vehicle studies, frequency analyses are essential in defining the vibration and noise sources and in evaluating the severity of the existing conditions. There are two methods available for making spectral analyses of tracked vehicle noise and vibrations. One involves analog analysis techniques which are made directly in the field with the vehicle in operation (ie, using wave analyzers, octave band analyzers, etc.). However, this method is inefficient and inaccurate because of the time required in making detailed

TABLE XIV. PROCEDURE FOR ANALYZING NOISE AND VIBRATION CHARACTERISTICS OF TRACKED VEHICLES

A DEVELOPMENT OF INSTRUMENTATION

1. Instrumentation system required that is capable of obtaining sound and vibration data which can be used for multiple frequency analyses (ie, magnetic tape recording system).

2. System required which can produce continuous records of vibration data as operating conditions vary (ie, light beam oscillograph or FM tape recording system).

3. Supporting instrumentation include sound level meter, sound and vibration calibrators, power supply system, and broad and narrow band noise and vibration analyzers.

B. STUDY OF NOISE AND VIBRATION CHARACTERISTICS EXISTING IN TRACKED VEHICLES TO DEVELOP AN UNDERSTANDING OF THE NATURE AND EXTENT OF THE PROBLEMS

1. Airborne Noise Studies

   a. Compare to criteria

   b. Look for unique sonic signatures or frequencies

   c. Evaluate existing noise levels as a function of operating conditions to determine further testing

2. Vibration Studies

   a. Look for prominent frequency components

   b. Evaluate the noise radiating potential of the vehicle panels by use of sound-power index

   c. Conduct impulse testing for measurement of the natural frequencies and inherent response of the various panels and radiators.

C. SYSTEMATIC TESTING FOR THE EVALUATION OF VIBRATION SOURCES AND TRANSMISSION PATHS

1. Measurement of sound and vibration at selected positions while operating at selected speeds under a variety of conditions.

TABLE XIV. (Cont'd)

a. Roads of different types of terrain; for road noise evaluation

b. Up and down hill; for engine, transmission, and track road noise studies

c. Same speed, different gear ranges; for additional engine and transmission load noise studies

d. Swimming; to show composite effects of liquid damping and road surface noise

e. Various track tensions; for studying effects of the track reactions and track resonances.

f. External muffler; additional studies of importance of radiated engine noise.

g. Vehicle on blocks; eliminating road-track reaction

h. Vehicle on blocks, only one track free to turn; vibration transmission studies.

i. Rubber idlers; study of vibration damping afforded by rubber instead of steel idlers

j. Towing with engine running; for evaluation of engine, transmission, and track load noise.

k. Towing with engine off; eliminating engine noise in addition to other aspects in (j).

l. Towing with tracks off, engine on, transmission in gear, showing effects of removing the track noise (track reactions with sprockets, idlers, and road wheels).

m. Towing with tracks off, engine on, transmission in neutral; for transmission noise studies.

n. Towing with tracks off, engine off, transmission in neutral; for additional engine noise studies.

o. Towing with tracks off on various terrain; for suspension system noise studies.

TABLE XIV. (Cont'd)

p. Standing still, tracks off, engine running, transmission in gear; additional studies of unloaded engine and transmission noise.

q. Standing still, engine running, transmission in neutral; studying the unloaded engine noise by itself.

r. Vehicle on blocks, additional source modification tests; for vibration transmission studies (tests directly applicable to M-113 type vehicle).

   (1) Rubber pads glued on sprocket teeth; studying effects of removing direct metal-to-metal contact between sprockets and tracks.

   (2) Metal sprocket toothed rings removed; studying effects of removing metal-to-metal contact between sprockets and tracks.

   (3) Rubber idlers; studying vibration damping effects afforded by rubber idlers relative to steel ones.

   (4) Rubber cushions removed from sprockets; evaluating effects produced by these cushions.

2. Impulse Testing Techniques

   a. Study relative coupling paths between vibration sources and noise radiators.

   b. Trace vibrations through the suspension system.

analyses, and the difficulty in maintaining the same operating conditions for the required long period of time. Therefore, it is highly probable that parts of such analyses will be made under operating and noise conditions different from those desired. The alternate method is to make brief permanent recordings of the desired data and then conduct the reduction of this data in the laboratory. This latter method is much more advantageous and convenient. The most suitable method for recording the noise data for subsequent analyses is through the use of a portable magnetic tape recorder. By making loops of the taped data, analyses can be conducted to the extent desired.

However, the use of a normal tape recording instrumentation system alone is not sufficient for making a thorough survey of tracked vehicle noise and vibrations. The tape recorder is best adapted for taking data at fixed operating conditions (i.e., constant speed, uniform road surface, etc.). If the operating conditions are allowed to vary while tape recordings are being made, the analyses of the data become complicated. Thus, for exploring the vibrations of panels and the existing noise levels as a function of operating conditions, another type of recorder such as a light beam oscillograph is advantageous.[9] With this type of instrument, it is possible to make continuous recordings of the vibration and noise levels as operating conditions are varied. These recordings of vibration can be readily analyzed for predominant driving frequencies and individual panel resonances. Because the observed sound spectra are much more complex than the vibration spectra, informative analyses of oscillograph recordings of sound are limited mainly to variations in the overall amplitude and possibly the identification of very prominent low frequency components. Because numerous channels are available on this type of recorder, however, measurements can be made at different positions at the same time. This allows studies of mode shape, phase differences, and vibration and radiated noise correlation to be made. Also, impulse testing for studying the characteristics of panels and for tracing vibration transmission paths can be conveniently made with an instrumentation system employing such an oscillograph. Through systems utilizing basic recorders of these two types, most of the phases of the noise and vibration measurement of tracked vehicles can be exploited.

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9 An alternate system would involve the use of multi-channel FM recorders. These units are useful from 0 cps to well above the audible range. Such recordings are more complex to analyze than the light beam oscillograph records, however, chiefly because visual observation is impossible and correlation of data to operating conditions is more difficult. In addition they are more difficult to operate and calibrate.

The recording instruments themselves, however, are only parts of the overall measuring systems. Other essential components include transducers, microphones, amplifiers, proper connecting cables, and the necessary power supplies. An important requirement of the vibration transducers, in addition to possessing a flat frequency response throughout the range that vibrations exist in tracked vehicles, is that their weight should be sufficiently light so that attaching them to various elements in the vehicle does not affect the vibration response of these elements. An example of the type of transducer that will give satisfactory results is the Endevco Model 2215C accelerometer. Any type possessing similar characteristics to this model should suffice. To amplify the signals picked up by the transducers, again, of fundamental importance is the frequency range under consideration which essentially starts at DC or 0 cps and extends throughout the audible range. Since most accelerometers require high impedance loads for maximum frequency response, the first stage of the amplifiers should include this feature. Solid state amplifiers with their compact sizes and simple power supply requirements are convenient.

Another factor for consideration in the measurement of vibration is that sound radiation from a panel is proportional to the panel velocity rather than acceleration. Thus, it is much more convenient to record the vibrations in terms of their velocities. If accelerometer transducers are used to pick up the signals, then the amplifier stage should have the capability of converting these signals to velocity signals.

To measure noise levels directly, the sound recording system should possess the same fundamental requirements as the vibration system. Again, a primary factor is the frequency range of interest. An example of the type of microphone which works satisfactorily under the existing conditions is the Shure 98B99 ceramic model. Figures 1 and 2 in Section II illustrate examples of the complete instrumentation systems.

Another instrument which is extremely useful in noise measurement studies is the sound level meter. This instrument provides a means for making simple absolute sound level readings directly in the field for studying the severity of the conditions and for comparing various test effects, thus laying the foundation for additional and more detailed testing. The sound level meter also provides a method for checking the accuracy of the other measuring techniques by comparing the levels measured with the meter to the overall levels recorded with the other methods.

Another essential part of any measuring technique is the calibration of the instrumentation. Calibration signals are necessary for determining the intensity of the measured noise and vibration levels and for conducting studies of a comparative nature such as those necessary to evaluate noise and vibration sources. To minimize the possibility of errors in the absolute readings from test to test, calibration signals should be recorded every time a particular instrumentation system is used.

Once the desired data has been obtained, the next step is the proper reduction of this data. Recording with an instrument such as a light beam oscillograph limits the available analyzing techniques. By visual observation of the recordings, it is difficult and time consuming to make analyses of complex spectra. As mentioned earlier, the oscillograph method of recording data is best suited for observing resonances and predominant driving frequencies as a function of operating conditions.

On the other hand, data recorded on magnetic tape can be used to make frequency analyses to any desired degree of detail. By the continuous replaying of loops made of the taped data, any particular test can be extended until the desired analysis is completed. One of the simplest, yet most informative, methods of studying noise and vibration problems is by octave-band analysis. This technique is effective in tracked vehicles because of the wide-band nature of the observed levels. However, narrow-band analyses should be made in addition to octave-band studies to identify possible prominent pure tones and to assist in the identification of vibration and noise generators by the correlation of observed frequency components with known driving frequencies. There are numerous types of narrow-band analyzers available for practically any extent of frequency resolution. For the M-113 tracked vehicle, it was observed that there was considerable modulation present in the noise and vibration levels under nearly all operating conditions; thus, minutely detailed analyses were not necessary or desirable. It is necessary to obtain a system of balance between the desired resolution and the repeatability of the analyses. For example, in the plotting of various vibration and sound spectra of the M-113 APC with a Panoramic subsonic analyzer, it was necessary to utilize a relatively broad resolution and a slow scan time in order to obtain meaningful and repeatable results. Thus, before successful instrumentation systems can be developed that can readily be applied to yield thorough evaluations of vibration sources, noise sources, and vibration transmission paths, it is essential that the nature and complexity of the existing vibrations and noise levels be understood first.

2. Testing Procedure

The most logical approach to the analysis of any tracked vehicle's vibration and noise characteristics is to begin by conducting extensive surveys of the noise levels that are created by the vehicle, both internal and external. The internal noise should be compared to hearing criteria so that the intensity and severity of the noise problems can be realized. The existence of possible pure tones or prominent frequencies should be studied, since the identification of such noise conditions can lead to the identification of specific vibration and noise sources. In addition, the variation of noise level with operating conditions should be investigated to determine the more offensive operating conditions and to define the variables upon which the radiated noise is most dependent. Specific parameters to be varied include vehicle speed, engine speed, road surface texture, road surface slope (ie, level, uphill, or downhill), and the position of the various hatches (either open or closed). From testing of this nature, the important operating parameters can be determined, and the foundation can be laid for further necessary tests for the evaluation of specific sources of noise and vibration.

In addition to preliminary noise studies, thorough vibration surveys are essential to formulate a complete working knowledge of the sound and vibration characteristics of any type of tracked vehicle. These studies should include the vibrations of the numerous panels and other elements of the vehicle as a function of operating conditions. From such surveys it is possible to evaluate the potential noise-radiating ability of the various panels, thus providing a means for comparing their importance as noise radiators. Of special importance in the study of panel vibrations is the realization that any particular panel can vibrate in many mode shapes, thereby complicating the problems of vibration measurement techniques and meaningful methods of predicting the radiating noise from panels. However, by strategically placing transducers at a number of positions on each panel, statistical averages can be calculated for the panel vibration velocities, and by applying the sound power index (u^2A) technique, it is possible to arrive at a logical estimate of the relative noise-radiating potential of the various panels.[10] In addition, the vibration characteristics of individual panels can be studied by conducting impulse testing on the panels. By measuring the vibrations of a panel excited by delivering an impulse to the panel, such features as the natural frequencies and inherent damping of the panel can be determined.

Once the general behavior of the existing noise and vibration characteristics is understood, the next step is to evolve a systematic test procedure that will yield the evaluation of specific vibration sources and the coupling paths between these sources and the previously identified noise radiators. The basic principle of such a procedure is to conduct various types of tests which will provide a means for directly comparing each source's contribution to the vehicle vibrations. Initially, several testing standards must be set. For example, the effect of road surface upon noise and vibration has previously

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[10] See Section III-C-4 for a more detailed discussion.

been evaluated, so it should be possible to select a basic test course for most of the remainder of the testing. A few different types of terrain are essential for a limited amount of further testing, but the majority of the work should be done on one representative test course. In addition, typical test speeds should be selected and used throughout the evaluation testing procedure. Also of importance is the selection of standard testing points. During the testing of the M-113 tracked vehicle, it was found that the simplest and one of the most informative techniques for comparing effects was to record the internal airborne noise at a centrally located position during the various tests. Since the radiated noise inside the vehicle is a summation of noise from all sources, then a measure of this noise provides a convenient means for studying the effects produced by the vehicle component vibrations in general, and of various testing designed to evaluate vibration sources. A centrally located measuring point is desired to minimize localized panel effects. In addition to the measurement of sound, it is beneficial to select a few positions to measure vibration during the preliminary evaluation tests. The vibration measurements are useful to verify the indications afforded by the noise data, for tracing vibration transmission, and to further define sources. For external noise source evaluation, the test position should be chosen sufficiently far from the vehicle so that preference is not given to any particular source.

The type of data reduction used for source evaluation is also extremely important. Total wide-band vibration amplitude measurement provides a condensed method of comparing the overall importance of vibration and noise generators. Another simple, yet informative, means of making comparisons is by octave-band analyses. However, some narrow-band analyses should be made in addition for the correlation of individual frequency components with the responsible sources.

The more detailed evaluation procedure involves a systematic testing sequence of selectively removing or altering various sources or coupling mechanisms. Table XIV contains a list of the numerous tests that can be conducted and the effects to be studied by each. It is noted that many of the tests demonstrate the same effect but are included for completeness and as alternate techniques for substantiating results. Most of the tests should be directly applicable to any type of tracked vehicle. Some of the techniques, however, might require special adaptation to fit other vehicles, since they were derived to study specific components of the M-113 APC.

In addition to the tests where vehicle operating conditions are varied, impulse testing techniques can be applied for the study of vibration transmission. Specifically, impulse tests are beneficial for evaluating the relative coupling between vibration sources and noise radiators and for tracing vibrations through complex systems such as vehicle suspension mechanisms.

In summary, although tracked vehicle noise and vibration characteristics are definitely complex in nature, it is possible through the application of a planned systematic test procedure to effectively resolve these complexities and analyze the existing conditions. Thus, reductions in the severity of these conditions should be forthcoming, since it can be readily determined where treatment techniques are required.

B. Suggested Areas For Further Work

As a result of the work conducted on defining instrumentation systems and measuring techniques, some definition has been made of the principal track and suspension system noise generators and transmission paths and of the radiating noise sources within the vehicle. While it has not been the object of this program to pursue abatement approaches applicable to these noise problems, some such work was conducted involving the modification of several of the primary noise generators and radiators. This work was performed chiefly to observe treatment effectiveness and to minimize certain offensive noise conditions in order that other primary generation and transmission tests could be more effectively pursued. In addition, some consideration was given to interior noise absorption treatments designed to reduce the reverberant level experienced within the vehicle.

As a result of this treatment work, as well as the work concerned with identifying the vibration generation sources in paths, several recommendations can be made relative to treatment techniques and further research which might be successful in reducing the overall noise problems associated with the M-113.

The noise field inside the vehicle is characterized by its high intensity, reverberant nature, and wide band frequency spectrum. As noted previously, the important radiating sources of this noise are a variety of responsive panels making up the interior surfaces and furnishings of the vehicle. Also it has been noted that the output of each of the individual sources is dependent upon operating conditions, and noise sources come and go as conditions vary.

It must be recognized that the treatment of no one source can be expected to greatly reduce noise conditions because of the multiplicity of comparable sources and the variation of source importance with operation conditions. Thus, while one source may be predominant (at a given condition or at all conditions) its complete removal will often reduce overall levels by only a few db (because of the nonlinear nature of the db). Even if the source treated has been contributing half of the total noise within the vehicle, its complete elimination would reduce interior noise levels only 3 db, and at many conditions the treatment would have virtually no effect. Hence, if this source were reduced over about 10 db, other noise sources would subsequently become important. It becomes apparent, therefore, that any treatment approach should consider all of the sources which contribute noise above the limits specified by applicable hearing damage or other criteria. In order to significantly reduce interior levels by modifying the individual sources, a rather comprehensive program would therefore be required.

While the program discussed herein was effective in defining many of the predominant vibration and noise generation sources, the exact mechanism of generation cannot be defined in all cases. For example, a more detailed analysis of the sprocket operation alone would be required before a complete evaluation can be given to all of the generation mechanisms occurring. Thus, from the recommendations given below, it is expected that a number of treatments will have to be incorporated before noise conditions in the vehicle are appreciably subdued. On the other hand, all suggested approaches will probably not be finally warranted for test, although each should be considered initially on the basis of need, feasibility, effect, and practicality.

1. Sprocket and Track Treatment and Redesign

Experimentation has shown that noise is generated by a number of mechanisms in the drive-sprocket wheels. For example, the impact of the teeth on the track contributes significantly to the high frequency content of the noise within the vehicle. On the other hand, complete removal of the sprocket teeth did not appreciably affect overall noise levels (running, on blocks), largely because the most significant contributor to overall noise is at the fundamental track shoe frequency which transmits directly through the rubber sprocket tire. In addition, some evidence was observed which indicated that impact of the track teeth on the idler and road wheels contributed some high frequency noise. On the basis of these observations, a number of possibilities exist for treatment or modification of the sprocket hub configuration, and of the other wheels, to minimize drive impact and impact generated by deformation of the track pitch as it progresses around the sprocket. While it is not certain that all of these approaches will prove successful, the potential of each appears to warrant consideration. In each case, an analysis should be made both of present noise considerations and previous work by others as to the feasibility and effectiveness of any considered redesign or treatment. If the potential for proposed treatment still appears good, work should proceed into the application and testing stages. The apparent possibilities for improved design include the following:

a. Redesign of sprocket teeth to provide more gradual change in track pitch and reduce impact. While much work has been done by others on tooth form and tooth material, it appears warranted to consider these and other facets (eg, use of individual teeth, etc.) from an impact and noise standpoint. Such work may prove effective in reducing much of the high frequency noise generated in the sprocket.

b. Alteration of the sprocket and/or hub pitch diameter, and alteration of the resilient material in the tire to reduce impact generation and transmission.

c. Vibration isolation of the sprocket disk from the hub to minimize impact transmissibility from teeth to hub and subsequently to drive shaft. This might also include consideration of individual teeth rather than teeth cut from a single disk.

d. Isolation of the sprocket bearing housing from the vehicle chassis. This would involve mechanically decoupling vibration generated by the sprocket from the remainder of the vehicle.

e. Consideration of perhaps ultimately evolving new drive techniques to eliminate the sprocket in its present form and new track designs. Since the track reactions are the primary offenders, effective track redesign techniques, if feasible, would be an efficient method of producing substantial results. Specific areas that might be considered include modification of the track pitch or eliminating pitch consideration by using a continuous track, reducing the mass of the track by redesign or by use of different materials, and improving the vibration isolation concepts of the present track.

Each of the approaches would require preliminary concept and design considerations before final recommendations could be made; however, such efforts should shed considerable insight as to the usefulness, feasibility and practicality of the approaches. A minimum design effort should be required for the most promising of the several approaches, ie, the isolation of the sprocket hub from the vehicle, and the teeth from the sprocket hub. This might require a new hub casting or modification of an existing one for insertion of resilient wafers or other isolating material. On the other hand, item "e" is one which would be considered in conjunction with work on the other four, and probably should not be considered as a separate item until, and if, definite ideas are generated.

2. <u>Idler Wheel and Suspension Isolation</u>

If reduction of sprocket-drive noise is achieved, results of the program indicate that isolation of the rear idler wheel will also be warranted to bring its noise level back into line (ie, at a newer lower level). It is probable that techniques similar to those of item "d" above could be used with a slight redesign, and perhaps a re-adoption of rubber tire idlers. The most obvious approach involves an insertion of a resilient gasket in the idler support system to mechanically isolate vibrations generated in the idler. Another possibility lies in modifying the design of the tension-adjust cylinder. Similarly, some improvement of road wheel suspension noise might become important if other sources are successfully treated. Consideration should be given to a means for isolating the torsion bar bearing and bracket system to minimize at least its high frequency noise conduction.

3. Internal Noise Treatment

During the course of the present program, rather crudely fabricated acoustical absorption and isolation techniques have been shown to significantly reduce the internal noise levels. These tests showed that when such treatments are properly designed and installed, they can serve effectively to reduce reverberant build-ups and overall levels within the vehicle. It is suggested, therefore, that these approaches be pursued and that resulting treatment designs be more carefully engineered both to optimize the acoustical effects and to make the treatments compatible in the material and construction requirements of the vehicle. While the crude tests have shown certain approaches to be effective, additional work will be required to evolve practical and effective designs. Specifically, this work should include at least the following:

a. Design, installation, and evaluation of new floor panel concepts to isolate solid-borne noise, providing increased panel damping, and providing high transmission loss characteristics for noise radiated from the vehicle bottom hull.

b. Design, install, and evaluate new engine-access panels to provide mechanical isolation from the support structure, increased panel damping, increased transmission loss, and perhaps increased acoustical absorption in both the personnel area and the engine space (ie, to minimize sound build-up in the engine compartment and subsequent radiation through the air inlet grill to the outside).

c. Additional acoustical absorption within the interior of the vehicle to reduce reverberant effects of sound build-up and speech interference.

d. Treatment of other interior noise radiating components, such as battery box, seat back rests, gas tank panels, etc., which have been shown to be offensive noise radiators under some operating conditions. This would employ vibration control, acoustical treatment, and component redesign techniques.

4. <u>Analysis, Redesign, and Evaluation of Engine Muffler and Air Intake Systems</u>

Tests are shown that the engine exhaust and ventilation noises are significant outside the vehicle and within the vehicle under some operating conditions. It is recommended therefore, that the muffler and exhaust system be redesigned in order to obtain greater overall attenuation and minimize system acoustical resonances and pass-bands. Similarly, an evaluation should be given to reducing the noise transmitted from the engine compartment to the outside. It is probable that treatments can be evolved using lined-duct and lined-bend techniques to appreciably reduce the transmitted noise from the engine itself.

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